INDUSTRIAL TECHNOLOGY CENTRE/MANITOBA RESEARCH COUNCIL

1329 Niakwa Road East, Winnipeg, Manitoba R2J 3T4 Telephone (204) 945-6000

IDENTIFICATION OF DEFECTS
in
Cast Steel Components
by
VIBRATION ANALYSIS

COPY

Final Report
by

Industrial Technology Centre
Division of Manitoba Research Council
1329 Niakwa Road East
Winnipeg, Manitoba
R2J 3T4

Work on the project was conducted under the auspices of
the Canada Centre for Mineral and Energy Technology, Energy,
Energy, Mines and Resources, Canada.

DSS File Number: 14SQ.23440-5-9188
DSS Contract Serial Number: ISQ85-00202

Canmet Project #U-M-542
Canmet Scientific Authority: Mr. V. Caron
May 19, 1987

MICROMEDIA

Project #01377

Summary

A vibration response technique as a potential non-destructive test method to identify defects in cast components has been evaluated. After a review of existing literature, extensive efforts were devoted to develop proper techniques and instrumentation which were effective and simple to use.

Of the three dynamic characteristics; mode shape, damping, and natural frequency, the latter was selected to be most suitable for this study. The effect of different variables such as casting dimensions, temperature and defects on natural frequency were also investigated.

The merits and simplicity of the method were verified in a case study using shell molded cast chisel plow teeth. The shell molded chisel plow teeth showed a significant decrease in natural frequency due to defects such as porosity and inclusions. Good correlation between radiographic quality and natural frequency was also witnessed.

Résumé

Une méthode employant des réponses en fréquence a été vérifiée comme une épreuve de vérification non-destructrice pour déterminer les loups d'une pièce coulée. Après avoir fait une bibliographie complète, on a fait un effort spécial pour retrouver la meilleure façon de faire cette épreuve avec un appareillage simple et efficace.

A partir de ces trois traits caractéristiques, soit la forme modulaire, l'amortissement, et la fréquence naturelle, ce dernier a été choisi comme etant le trait qui offre les plus grands avantages. La conséquence des paramètres différents tel que la dimension du moulage, la température, et les défauts sur la fréquence naturelle ont été étudiés.

Les qualités intrinsèques et simplistes de la méthode ont été vérifiées dans une épreuve employant la charrue dentée moulée. Ce moulage a démontré qu'il y avait une décroissance de fréquence naturelle grâce à les défauts tel que la porosité et les inclusions. On a observé une bonne corrélation entre la qualité radiographique et la fréquence naturelle.

Table of Contents

	Page
SUMMARY	i
List of Tables	iv
List of Figures	v
1.0 INTRODUCTION	1
2.0 LITERATURE REVIEW	2
3.0 TEST METHOD DEVELOPMENT	4
3.1 Review of Vibration Response of Structures	4
3.2 Measurement of Frequency Response	5
3.2.1 FFT Analysis Resolution	7
3.2.2 Determination of Modal Parameters	8
3.3 Dynamic Characteristics Monitored for NDT Method	9
3.3.1 Natural Frequency	9
3.3.2 Mode Shape	9
3.3.3 Damping Factor	11
3.4 Selection of Response Transducer	12
3.4.1 Contact Transducers	12
3.4.2 Non-Contact Transducers	13
3.4.2.1 PROXIMITY TRANSDUCERS	14
3.4.2.2 OPTICAL SYSTEMS	14
3.4.2.3 MICROPHONE	15
3.5 Temperature Effects	16
3.5.1 Axial Vibration of Uniform Bars	17
3.5.2 Uniform Beams in Flexural Vibration	18
3.6 Dimensional Tolerance Effects	20
3.6.1 Axial Vibration of Uniform Bars	21
3.6.2 Uniform Rectangular Bars in Flexure	22
3.7 Test Method Development Summary	24
4.0 CASE STUDY	25
4.1 Sample Description	25
4.2 Procedure	26

Table of Contents - continued

	Page
4.3 Results	26
4.4 Discussion	26
5. CONCLUSIONS	27
REFERENCES	28

List of Tables

1. Shift in Natural Frequencies Due to Changes in Dimensions.

2. Measured Data Representing Natural Frequencies, Weight and Dimensions of the Knee-Joint Casting.

3. Statistical Data Related to Natural Frequencies, Weight and Dimensions for Knee-Joint Casting.

4. Radiographic Level of Each Chisel Plow Tooth Casting.

5. Measured Data Representing Natural Frequencies, Weight and Dimension of Plow Teeth Casting.

6. Statistical Data Related to Natural Frequencies, Weight and Dimensions of Plow Teeth Casting.

List of Figures

1. Graphic representation of a computer model of a rectangular steel plate.

2. Frequency response function of displacement at G1 on plate due to force applied at G5.

3. Deformed geometry plot of mode shape of second mode of vibration of steel plate.

4. Plot of FRF modulus (upper curve) and phase (lower curve) for various values of the modal damping factor.

5. (a) Graph of force versus time for typical impact excitation. (b) Energy spectral density (spectrum) of the impact force.

6. (a) Graph of acceleration versus time for typical response on casting due to impact. (b) Energy spectral density (spectrum) of the acceleration response.

7. Typical FRF measured on a casting using the impact test method.

8. Typical impact test hammer.

9. Graph of force versus time for three different impacts (upper curve) and the Fourier transforms of the same impact forces.

10. Impact test of a steel casting with baseband analysis frequency of 4.0 kHz; impact force spectrum (upper left), acceleration response spectrum (lower right), and FRF (bottom).

11. Impact test of a steel casting with baseband analysis frequency of 4.0 kHz and zoom factor of 16; impact force spectrum (upper left), acceleration response spectrum (lower right), and FRF (bottom).

12. Impact test of a steel casting with baseband analysis frequency of 4.0 kHz and zoom factor of 128; impact force spectrum (upper left) acceleration response spectrum (lower right), and FRF (bottom).

List of Figures - continued

13. (a) Impact response spectrum measured on a casting using an accelerometer (upper graph). (b) Impact response spectrum of the same casting measured by a microphone in the near field.

14. Graph showing the effect of ambient temperature on three natural frequencies of vibration of a steel casting.

15. Graphs of relative change in natural frequency of axial vibration of uniform bars and flexural vibration of uniform beams versus dimension and casting weight for tolerance grades T3 (a) and T7 (b).

16. Investment cast knee joint artificial implant component.

17. Histograms showing probability of occurrence versus first (a) and second (b) natural frequency of knee joint casting.

18. Vibration response testing set up.

19. Shell mold cast chisel plow tooth casting.

20. a - chisel plow tooth radiograph - Level I.
b - chisel plow tooth radiograph - Level II.
c - chisel plow tooth radiograph - Level III.
d - chisel plow tooth radiograph - Level IV.

21. Histogram showing probability of occurrence versus first (a) and second (b) natural frequency of chisel plot teeth.

22. Same as Figure 21(a) except radiographic level of each group indicated above them.

23. Variation of first natural frequency with radiographic level of chisel plow teeth.

1.0 INTRODUCTION

Due to the complexity of modern steel casting processes, cast products often contain material flaws. These flaws can be in the form of porosity, inclusions or cracks induced by subsequent heat treatment. Since the strength and performance of the product is reduced by flaws, premature failure having disastrous results may occur if the defects are left uncorrected.

The first step in correcting manufacturing defects is to detect the presence of flaws. This can be a difficult task since many flaws occur below the surface of the part making their detection by the naked eye impossible. Ideally, techniques to detect flaws should be non-destructive, quick and easy to perform and interpret, and easily integrated into production processes.

Currently, six types of non-destructive evaluation (NDE) techniques have found broad use in industry. These NDE techniques are radiography, ultrasonics, eddy current, acoustic emission, liquid penetrant, and magnetic flux. Of these, only ultrasonic and radiography are capable of detecting flaws substantially below the surface of the part. Although well proven, both of these techniques have limitations.

Ultrasonic NDE is a very localized technique which can only evaluate a small amount of material at one time. Therefore, a non-destructive evaluation by ultrasonics requires several tests to evaluate the entire part. Radiography has the limitations that it is time consuming, and unable to detect discontinuities such as cracks which are more or less perpendicular to the radiation beam. It is also noted that neither technique performs well in the vicinity of welds or abrupt changes in geometry.

The non-destructive evaluation technique which is the subject of this study is known as "vibration response" or "sonic resonance". The technique involves measuring the dynamic response at one point on a part, due to excitation applied to some other point on the part.

It is well known that all structures have an infinite number of natural frequencies of vibration. The natural frequencies are characteristic values which depend on the structure's geometry, and material properties.

When a part is excited with a dynamic force at one of its natural frequencies, it absorbs energy very readily and vibrates with large amplitudes. Furthermore, if the part is excited with an impulsive force, it is excited at all frequencies over a broad frequency band simultaneously. If the excitation force and the response at a point on the part are monitored, Fourier transform techniques may be used to determine a frequency response function (FRF). This function displays

how the relative response of the part varies with excitation frequency and constitutes a signature of the structure. Any differences between the frequency response function of two geometrically similar parts are due to differences in their mechanical properties or mechanical integrity.

The objectives of this study are three fold:

1) To determine the feasibility of using vibration response analysis as a non-destructive evaluation method to identify the presence of defects in steel castings.

2) To refine the response analysis technique by determining the optimum type of response transducer, excitation method, and analysis method.

3) To demonstrate the simplicity and completeness of vibration response analysis as a non-destructive evaluation technique for quality control in steel casting production.

To achieve these objectives a study consisting of: a review of available literature, an evaluation of the influences of various apparatus, specimen properties, and environmental conditions; and the application of the method to shell mold cast parts was pursued.

2.0 LITERATURE REVIEW

Vibration response or sonic testing of components for detecting quality or soundness has been used for centuries. Striking an object causes it to ring with a characteristic sound, and the presence of defects such as cracks produces an "abnormal" ringing sound. The success of this technique is evident in the case of chinawares and potteries. Engineers and scientists have attempted to develop vibration response for engineering components over the past two decades. In the following some of their published findings will be reviewed.

Resonant frequency and damping properties have been extensively studied for use in assessing quality of cast-iron components. The British Cast-Iron Research Association (BCIRA) investigated the vibration response to predict mechanical properties of ductile iron (1-5) and confirmed that resonant frequency or ultrasonic velocity measurements can be used to predict, within close limits, the effect of small changes in graphite form or nodularity on the tensile strength. The matrix structure, shape and size of graphite also influenced the resonant frequency and good correlation was established with graphite parameters.

Rhoda (6) investigated sonic testing of gray iron sleeve castings to predict tensile strength of the material. Using a simple vibration generator and frequency meter he established a linear relationship between tensile strength (graphite flake morphology) and resonant frequency.

Kovacs (7) at Ford Motor Company demonstrated the use of vibration response testing in checking the quality of ductile iron components like crankshafts, connecting rods and gear case covers. Using impact signal generation as well as forced sinusoidal signal input and analyzing the frequency spectrum, he was able to predict nodularity in the casting along with yield and tensile strengths. Common defects such as shrinkage porosity, inclusions and exaggerated nodule growth lowered the resonant frequency.

For gray-iron castings, Adams et al (8) used dynamic torsional testing to determine resonant frequency and dynamic modulus. Changes in size and shape of graphite flakes correlated well with the modulus parameter.

Using vibration characteristics to determine integrity of components has been used on materials other than cast-iron. Decay of wooden poles used in electric power transmission has been studied by Breeze et al(9) and Bodi et al (10). They demonstrated that sonic testing across the diameter of poles and subjectively analyzing the sound permitted them to determine the strength of the wooden poles. For ceramic steel-casting plugs, Blurshtein et al (11) was able to detect thermal shock resistance as well as presence of cracks in components by measuring resonant vibration frequency. Sonic pulse-echo technique through impact of a hammer has been shown by Alexander (12) to detect voids and discontinuities in concrete slabs. For plastics, Reneker (13) demonstrated the use of modern tools such as fast Fourier transforms (FFT) analysis and frequency spectrum analysis for detecting flaws such as cracks and voids which alter the spectrum in terms of shifts in peaks or amplitudes.

Vibration signature analysis has been successfully used for years as a preventive maintenance tool in rotating machinery applications (14). The forcing function consists of the operating forces of the machinery whose vibration signature is monitored periodically. Any deviations from the baseline signature indicate potential problems such as bearing failures, shaft imbalance, misalignment, etc. Baird (15) demonstrated this for aircraft type fans and pumps where scratches in the bearing surfaces could be detected using frequency spectra.

Artificial excitation through impact hammer and FFT analysis techniques to analyze characteristic frequency response spectra of components has been used by Smiley (16) to detect weld defects in welded truck wheels and automotive frames. Adam et al (17) were able to detect the location of cracks in structures through correlation of experimental frequency analysis with extensive mathematical models. Harris (18) indicated that vibration analysis could detect gross damages to the offshore structures, the loading being induced by wind, waves, and currents. Similar techniques have been extended to detect rot in apples (19) and damage to composites (20).

3.0 TEST METHOD DEVELOPMENT

The potential nondestructive test method under investigation involves using an anomaly in the vibration response of a casting, relative to established norms, to detect the presence of material defects. The following conditions are necessary for this method to be successful:

1. There must be a method of measuring vibration response of castings which is simple and quick to perform, but yields consistent, repeatable results.

2. Anomalies in the vibration response of castings due to material defects must be greater in magnitude than differences in vibration response due to normal variation in the casting production process.

3.1 Review of Vibration Response of Structures

Like any elastic structure, castings have characteristic dynamic properties which determine how they vibrate in response to externally applied forces. These properties are called normal modes of vibration.

Normal modes of vibration occur at certain "natural frequencies". When a structure is excited with a force at one of its natural frequencies, it absorbs energy readily and vibrates with large amplitudes in a characteristic pattern or deformation shape, called a mode shape. This is a manifestation of the energy trapped within the boundaries of the structure which can only be dissipated slowly.

Normal modes of vibration are fundamental properties of a structure which are independent of how the structure is excited. When we look for anomalies in the vibration response of one casting compared to a control group, we are really looking for differences in the normal modes of vibration. Variations in these fundamental properties can only be due to differences in geometry or material properties. Since a production run of castings is geometrically identical, within certain tolerance limits, the vibration response method attempts to use variations in the normal mode properties of castings to identify variations in their material properties, i.e. defects.

In section 1.0, the concept of a Frequency Response Function (FRF) was introduced. A FRF describes the dynamic response at one point on a structure (or component) due to excitation applied at another point. For example, consider the computer model of a steel plate shown in Figure 1. Figure 2 is a FRF of displacement response between locations G1 and G5 on the plate. The FRF is shown in the form of a Bode plot. The lower plot called the modulus, represents the magnitude of response at point G1 on the plate per unit force applied at point G5, as a function of excitation frequency. The upper curve represents the phase angle between the applied force and the response.

An FRF of a structure contains information about its normal modes of vibration. In the FRF in Figure 2, four peaks in the magnitude curve can be clearly identified. Associated with each of the peaks, the phase changes rapidly by 180 degrees. These indicate the presence of normal modes of vibration. The frequencies at which the peaks occur are the natural frequencies of vibration.

Note that the FRF magnitude is plotted on logarithmic curve, and that the magnitude at the peaks is much greater than at frequencies away from the peaks. This indicates that the response per unit force is very high at a natural frequency, relative to other frequencies.

Another property of a normal mode is the mode shape. This is the characteristic deformation pattern in which the structure vibrates if excited at one of its natural frequencies. Each normal mode has its own mode shape. Figure 3 is a plot of a typical mode shape for the plate model. This mode shape corresponds to the second mode identifiable in Figure 2.

At a natural frequency of vibration, the magnitude of the FRF for a point on the structure corresponds to the amplitude of the mode shape at that point, related to other points. Similarly, the phase of the FRF corresponds to the phase of the mode shape relative to other points.

A third property of normal mode of vibration is the modal damping factor. The damping factor indicates the level of damping present. If no damping was present (damping factor = 0) the response of a structure excited at its natural frequency would be infinite. In practice, all structures have some damping. Figure 4 shows the effect of the modal damping factor on the frequency response of a structure.

These three properties of normal modes, the natural frequency, the mode shape, and the modal damping factor are called "modal parameters." They can be determined experimentally for most structures, and in fact a well-developed branch of applied mechanics known as "Experimental Modal Analysis" exists, which is principally concerned with the science of measuring modal parameters.

Since the vibration response method will attempt to use variations in the modes of vibration of castings to detect flaws, an important question to be answered is, "Which modal parameters should be monitored?" Before this question can be answered, it is necessary to consider how the frequency response of a structure, and hence its modal parameters, are measured.

3.2 Measurement of Frequency Response

Experimental modal analysis, or modal testing, began in the 1950's, during the early years of the space program. At that time, only one

method of experimentally determining the modal parameters of a structure was known. Called sine testing, this method involves exciting one mode of a structure at a time with a sinusoidal force. A mechanical shaker is attached to the structure and then driven with a sinusoidal signal. The frequency of the driving signal is then adjusted to coincide with one of the structure's natural frequencies. At a natural frequency, the motion will be the mode shape of the mode being excited. Response transducers are used to measure the amplitude and phase of the motion at points of interest on the structure. Modal damping is measured by switching off the shaker and measuring the decay rate of the sinusoidal motion in the structure. Sine testing is time consuming and requires an elaborate test setup.

With the discover of the Fast Fourier Transform (FFT) in the mid-1960's, and the advent of FFT analyzers soon after, a whole new approach to modal testing was born. The ability of the FFT analyzer to rapidly determine the frequency content of measured signals means that a structure can be excited over a broad band of frequencies simultaneously, and consequently, many normal modes can be excited and analyzed at the same time. With this method, known as the Transfer Function Method, signals from the force and response transducers are input directly to the FFT analyzer which rapidly determines the frequency response function from the measured signals. The transfer function method is much faster and easier to perform than the sine testing method.

To use the transfer function method to advantage, the structure must be excited over a range of frequencies simultaneously. One way of providing such excitation is with a mechanical shaker driven by a signal which contains energy at all frequencies in the band of interest, for example, random noise. This excitation method, although faster and simpler than the sine testing method, still requires a fairly elaborate setup, and is included here only for completeness.

Another way of providing broad band excitation to a structure is with impulse testing. When a structure is struck with an impactor (e.g. a hammer) an impulsive force is applied to the structure. An impulse contains energy at all frequencies within a certain range, and hence excites all modes within this range simultaneously. Impulse testing is the method of interest for this study because it satisfies the condition of being quick and easy to perform, with reasonable consistency of results.

Figure 5 (a) shows a typical curve of the applied force versus time when a casting is struck. When the Fourier Transform of the force-time curve is taken, the resulting frequency domain curve (Figure 5[B]), contains energy at all frequencies up to a certain frequency. Figure 6 [a] shows the measured time domain acceleration response of the casting to the applied impulse, at a certain point. The peaks in the

acceleration-frequency curve (Figure 6 [b]) occur at natural frequencies of the casting, indicating several normal modes of the casting were excited. The FFT analyzer computes the frequency response function of the casting between the impact point and the response point by dividing the Fourier transform of the response by the Fourier transform of the excitation, Figure 7. A FRF such as that shown in the figure is computed by the FFT analyzer in only a few seconds.

Usually, for impulse testing the impactor consists of a special hammer with a load cell or force gauge built into the hammer tip (Figure 8). The maximum frequency range at which a structure can be excited by an impactor is a function of the shape and duration of the force-time pulse. This, in turn is a function of the contact stiffness between the impact tip and the structure, and the mass of the impactor. Since these variables can be controlled to some degree by varying the mass of the hammer and the hardness of the tip, most impulse test hammers come with interchangeable tips and a means of varying the mass of the hammer head. Figure 9 shows several force-time pulses and their Fourier transforms.

The impulse method is a quick means of experimentally determining frequency response functions of structures. Since the FRF's contain information about the structure's modal parameters, or dynamic characteristics, it is this method which will be the subject of this study.

3.2.1 FFT Analysis Resolution:

The Fast Fourier Transform is an algorithm implemented within FFT analyzers to compute the Discrete Fourier Transform (DFT). The analyzer digitizes a continuous input signal and approximates it as an integer number of points in the time domain. The DFT then transforms this block of time domain data into an integer number of discrete spectral values. The number of time domain and frequency domain points sampled varies from analyzer to analyzer, but typical values are 1024 discrete time domain samples, and 400 discrete frequency domain samples.

Because it is a discrete rather than continuous Fourier Transform which is computed, the higher the frequency bandwidth analyzed, the lower the frequency resolution. For example, a 400-point FFT analyzer would have a frequency resolution of 0.25 Hz on a 100 Hz analysis bandwidth, but only 25 Hz resolution on a 10 kHz analysis bandwidth.

The frequency domain resolution of an FFT analyzer may be improved with a mathematical transform known as the Zoom transform (21). This transform improves the resolution by providing the same number

of discrete spectral values within a sub-band of the frequency analysis baseband. However, it does so at the expense of having to analyze proportionally more time domain data. For example, to increase the frequency domain resolution by a factor of four, four times as much time domain data must be included in the analysis.

3.2.2 Determination of Modal Parameters

To determine the three modal parameters, natural frequency, mode shape, and modal damping factor, the following analysis of the measured frequency response functions is required.

1. Natural Frequency:

   Determination of the natural frequencies of a structure from the measured FRF's is a relatively simple task. The natural frequency is a global property of the structure. That is, a natural frequency remains constant, irrespective of where on the structure the impact is applied, and the response measured. The natural frequency may be found by simply noting the frequency at which peaks in the FRF magnitude occur. This assumes that the natural frequency occurs exactly at one of the discrete spectral values computed by the FFT. If greater accuracy is required, to determine a natural frequency which occurs between two discrete spectral values, a curve fitting procedure may be used.

   Since the mathematical form of a FRF is known, a curve may be fit to the discrete experimental data in the region of a natural frequency, to interpolate between discrete spectral values.

2. Mode Shape:

   To determine mode shapes experimentally, FRF's must be measured between a reference point on the structure, and each point at which the mode shape value is required. Mode shape values can be "peak picked", i.e. the magnitude and phase of the FRF picked manually. If more accuracy is required, curve fitting schemes can be used to interpolate between discrete spectral values.

3. Modal Damping Factor:

   Damping is also a global property of a structure. Two methods exist for determining modal damping factors from FRF's. The first involves curve fitting. The known mathematical form of the FRF is fit to the experimentally derived function, and in so doing an estimate of the modal damping factor becomes available. The second method is called the "half power bandwidth" method. this method estimates the modal damping factor from the frequency bandwidth between points immediately above and below the natural frequency on the FRF modulus curve

at which the magnitude is 3 dB below the magnitude of the peak. The accuracy of both these methods is dependent on the frequency resolution of the measured FRF.

3.3 Dynamic Characteristics Monitored for NDT Method

As discussed, castings, like all elastic structures, have certain fundamental dynamic characteristics called normal modes. Normal modes can be described completely by three properties, natural frequency, mode shape, and modal damping factor. Furthermore, the technology to experimentally determine the normal modes of structures is well developed. We shall now consider which of these modal parameters should be monitored for the vibration response NDT method. This will be influenced by 1) how sensitive each of the parameters is to material defects, and 2) how easy it is to measure each parameter.

3.3.1 Natural Frequency

As noted previously, determination of natural frequencies of vibration from measured vibraton response is fairly simple. Furthermore, as noted in the literature review, it has been demonstrated for other applications that natural frequencies of structures are sensitive to the presence of material defects. Therefore, it was concluded that natural frequency should definitely be studied in this project.

Note that the excitation spectrum from a typical force impulse (Figure 5[b]) is more or less flat within its useable frequency range. Because of this, the modulus of the FRF (Figure 7) has the same basic form as the response spectrum (Figure 6[b].) This suggests that if "peak picking" was the method used to estimate the natural frequency, the response spectrum alone would provide as good an estimate of the natural frequency as the FRF modulus. Therefore, if natural frequency was the only modal parameter monitored for NDT, the impulse force would not have to be measured. This would simplify both the instrumentation and the analysis procedure. However, due to the resolution problems cited, it would probably be necessary to use the zoom transform.

3.3.2 Mode Shape

As described, determining the mode shapes of a structure experimentally requires the measurement of many FRF's, and substantial data analysis. Therefore, it is not practical to attempt to detect flaws in production components by measuring the detailed mode shape. However, monitoring changes in the magnitude of a single FRF at a natural frequency may be warranted. This is

equivalent to monitoring a change in the mode shape at a single point on the structure, and for this reason it is included under this heading.

Given that we require a parameter that is sensitive to the presence of material defects, but quick and simple to measure, the FRF magnitude (or mode shape) has several disadvantages, as follows:

1. It is not a global parameter. Intuitively, one would expect the FRF magnitude to be affected by the presence of material defects more in the region of high modal response on the structure, than in areas of low modal response. This means that whether or not defects could be detected might depend on where the FRF was measured.

2. Test setup would be more difficult. To ensure consistency of results, the operator would have to ensure that exactly the same points were used for impact and response locations from casting to casting.

3. Frequency resolution could be a problem. Recall that the modulus of the FRF represents the magnitude of the response per unit excitation force between two points on a structure. It is determined by dividing the response spectrum by the excitation spectrum. When impulse excitation is used, problems can occur if the zoom transform is used to increase the frequency resolution.

   For impulse testing, the force signal is usually very short in duration compared to the length of the data block included in the frequency analysis. In a sense, the energy in the force pulse is averaged over the entire data block. Although most of the data block is theoretically void of any signal (zero amplitude), in practice, the data contains low level noise, due to the cumulative noise effects of the instrumentation system. For baseband analysis at high frequencies, say greater than 1 kHz, there is usually sufficient energy contained in the force pulse to ensure a relatively noise free spectrum. However, when the zoom transform is used, the energy contained in the force pulse is effectively averaged over several data blocks. As the zoom expansion factor increases, the energy content of the noise in the signal approaches the energy content of the signal itself, and the force spectrum becomes progressively more noisy.

   This phenomenon is illustrated in Figures 10 through 12, which are graphs of impulse test measurements on a steel casting with three different zoom expansion factors. In each figure, the spectrum of the impulse force is shown in the upper left, the response spectrum in the upper right, and the FRF at the bottom of the page. Figure 10 is for a baseband measurement from 0 to 4 kHz (resolution = 10.0 Hz, with no zoom expansion. Each of the curves in Figure 10

is relatively noise free. However, because the FRF is obtained by dividing the response spectrum by the force spectrum, it also contains considerable noise. Figure 12 is the same measurement with a zoom expansion factor of 128 (resolution = .078 Hz). Note that the response spectrum still has a clearly discernible peak, but the force spectrum is so noisy that the peak in the FRF is almost unrecognizable.

Clearly, when the zoom transform is used to achieve greater frequency resolution, the accuracy of the FRF magnitude becomes questionable. Since it is expected that the NDT method would require frequency resolution higher than available in a baseband measurement, it is doubtful that the FRF peak amplitude would be useful as a indicator of material defects.

Due to the above limitations, it was decided that changes in the peak magnitude of FRF (or mode shape) would not be considered further in this project as an indicator of the presence of material defects.

3.3.3 Damping Factor

Intuitively, one would expect that the damping factor would be a good parameter to monitor for the vibration response NDT method, since the presence of defects in a structure, particularly, cracks, would probably increase its tendency to dissipate energy through damping. While this may be true, because the modal damping factor of steel castings is so small, it is impractical to consider measuring changes in this parameter as an indicator of material defects. Consider the following passage from a reference text on vibration (18):

> "Structural metals and alloys usually have very low damping, except for specifically developed high damping materials. The primary mechanism for damping in metals below the cyclic stress sensitivity limit is the energy associated with dislocation motion back and forth across the grain boundaries. This effect is so remarkably small that values of the loss factor for steel and duralumin greater than 0.0001 at stresses less than 1/4 of yield are suspect as being influenced by the test procedure."

> "Since the energy loss due to cyclic straining in most metals is very much smaller than that due to relative motion of the parts, the exact value of the damping factor for metals is seldom of any practical interest in design."

In other words, the modal damping factors for solid metal objects such as steel castings are so small that even for well-designed

and carefully executed experiments under controlled laboratory conditions, the damping caused by the apparatus is of the same order of magnitude as the damping factor being measured. Since this violates the condition that the test method be quick and simple, the concept of using measured differences in modal damping factors as an indicator of the presence of material defects was not pursued further.

Therefore, it was decided that of the three modal parameters which describe a structure's normal modes of vibration, it would only be practical to monitor natural frequencies, for the purpose of this study. Furthermore, since the natural frequencies of a structure can be determined from the vibration response spectrum directly, without the need to measure the impulse force, it was decided that no further effort should be expended on the measurement of the impulse force.

3.4 Selection of Response Transducer

For the purpose of this study, a vibration response transducer should have the following characteristics:

1. It should faithfully reproduce the frequencies at which a structure vibrates.

2. It should have both a wide frequency range and a wide dynamic range.

3. It should be simple to set up, and its accuracy should be relatively insensitive to how it is set up.

3.4.1 Contact Transducers

Contact transducers generally consist of a frame or case containing a seismic mass (a mass connected to the transducer casing through a spring element.) The transducer frame is fastened to the structure, and as it moves with the vibration of the structure, the motion of the seismic mass relative to the frame is sensed. Accelerometers and velocity transducers which rely on this principle are commercially available. Accelerometers sense the force in the spring element, which is equivalent to the acceleration of the transducer frame. Velocity transducers sense the velocity of the seismic mass relative to the transducer frame.

Velocity transducers were not considered for this study, since they are usually quite large, and their use would be restricted to relatively large cast components.

Accelerometers are widely used as vibration transducers. They have sufficient dynamic range for the purposes of this study.

The frequency range of accelerometers is governed by the size of the seismic mass and stiffness of the spring element used. The seismic mass and spring element themselves are an elastic structure, and, therefore, have their own natural frequency of vibration. The useable frequency range of an accelerometer is usually limited to 1/4 of its natural frequency. Generally, the smaller the accelerometer, the higher is its useable frequency range. Accelerometers weighing as little as 0.5 grams with a useful frequency range of 25 kHz are commercially available. However, in practice, to achieve such a high useable frequency range, special precautions must be taken when mounting the accelerometer to the structure. This violates the required condition that the transducer's accuracy be relatively sensitive to the test setup.

A second problem with accelerometers is an effect known as "mass loading." Mounting an accelerometer to a structure is equivalent to adding mass at a point. If the mass of the accelerometer is large enough relative to the mass of the structure, this can change the natural frequency of the structure. This in itself would not present a problem if the effect was consistent from casting to casting. However, for this to be the case, the same mounting point would have to be used consistently, since the mass loading effect would depend on where the mass is added to the structure. With the miniature accelerometers on the market today, mass loading would only be a problem with very small castings.

A third problem with accelerometers is a function of where the accelerometer is mounted in relation to the vibration mode shapes of the structure. In the mode shape of the steel plate shown in Figure 3, several points can be seen at which no motion occurs. These points are called node points. Node points exist for each mode of vibration. If the accelerometer were mounted to a node point of the structure for a particular mode of vibration, it would be impossible to detect the natural frequency of that mode on the basis of response alone. This means that if an accelerometer was to be used as the response transducer for this application, it would be necessary to try a few different mounting points on the structure, to ensure that node points of the modes of interest were avoided.

In summary, accelerometers have most of the characteristics listed above as required for this study, but special precautions would have to be followed in mounting them to the casting, and in choosing the point on the casting at which to mount them.

3.4.2 Non-Contact Transducers

Three type of non-contact transducers were considered for this study: proximity transducers, optical systems, and microphones. These are discussed below.

3.4.2.1 PROXIMITY TRANSDUCERS

Proximity transducers function by sensing the displacement of the moving surface of a metal object relative to the transducer. Various physical properties such as capacitance, mutual inductance, and eddy currents are used which serve to translate the gap between the object and the transducer to a voltage which can be sensed by a recording instrument.

Proximity transducers have several disadvantages for an application such as the present one. These are:

1. Since they measure relative motion, care must be taken to ensure that the transducer mount itself does not vibrate.

2. The working distance between the transducer and the object is quite small (in order of .050 inches). To maintain such a gap consistently, special fixtures would be required.

3. For the small displacement typical of the present application, a fairly smooth surface in the casting would be required. This would require special surface preparation before the test.

Due to these limitations, proximity transducers were not considered further for this study.

3.4.2.2 OPTICAL SYSTEMS

Two optical systems which measure vibration of surfaces without touching them are commercially available. These are a laser doppler vibrometer, and an optical displacement follower.

The laser doppler vibrometer works on the principle of the Doppler shift. A laser beam is reflected off the vibrating surface. The reflected beam is compared to the original laser beam, and the Doppler shift of the reflected wave is used to determine the velocity of the vibrating surface. This instrument has the capability to measure extremely low vibration levels over a very wide frequency range. It would probably be quite suitable for the present application, but was not considered because of its high cost (greater than $30,000.)

The optical displacement follower electronically tracks the motion of a discontinuity in the optical image formed by light reflected from or emitted by a moving object. It can measure extremely small vibration displacements over a wide frequency range. Like the laser doppler vibrometer, it was not considered for this application due to its high cost.

A related system which uses optical fibres for optical displacement measurement exists, called the MTI Phonic Sensor. This system has the same limitations as other proximity transducers, and was not considered for this reason.

3.4.2.3 MICROPHONE

A microphone was also considered as a non-contact response transducer for this study. When a casting is struck with a hammer, this excites the casting and causes it to vibrate at one or more of its natural frequencies of vibration. This sets up an acoustic wave in the air in contact with the casting which is very similar in frequency content to the vibration of the casting. It is this acoustic waveform which impinges on the human ear and causes the perception of the characteristic ringing sound emanating from the casting. The human ear responds to the acoustic waveform and relays information about the frequency content of the sound to the brain. A microphone in the near field of the casting responds to the acoustic waveform in a similar fashion to the human ear. Thus, a microphone can be used in conjunction with an FFT analyzer to quantify the frequency content of the sound emitted from the casting when it is struck.

Figure 13[a] is the response of a steel casting to a hammer blow, as measured with an accelerometer in the near field of the casting. For comparison, the response of the casting to the same excitation, as measured by a microphone mounted to its surface, is shown in Figure 13[b]. Note that at frequencies away from the natural frequencies, the acoustic response spectrum is much more noisy than the acceleration response spectrum. This is expected, since the microphone is much more susceptible to ambient acoustic noise than the accelerometer is to ambient vibration. However, at the natural frequencies , the acoustic response spectrum has well-defined peaks which are 20 to 40 dB above the background noise. Furthermore, the peaks in the acoustic response spectrum occur at exactly the same frequencies as the peaks in the acceleration spectrum.

In impact tests of various steel castings, a microphone in the near field of the casting was found to give very consistent results in terms of the natural frequencies indicated. This was true regardless of the position of the microphone relative to the casting. In other words, the accuracy of the microphone in sensing the frequency of the casting was insensitive to how it was set up.

The frequency range of a good quality microphone is the same as that of the human ear, about 100 Hz to 20 kHz. This is adequate for the present application, since in most cases, frequencies higher than about 10 to 15 kHz are difficult to excite using simple impact testing.

The dynamic range of a microphone is dependent on the electronic signal conditioner used with it. Generally, instrumentation used for acoustic measurements is quite adequate in dynamic range for the present application.

Since it met all three of the criteria required of a response transducer for this application, a microphone located in the near field of the casting was chosen for the response transducer. It was decided that a sound level meter would be used, since this instrument provides the necessary high quality microphone, signal conditioners and amplifier, in a compact package. The analog output from the sound level meter would serve as the input to the FFT analyzer for frequency analysis.

3.5 Temperature Effects

The vibration response NDT method will attempt to identify defects in steel castings by measuring differences in the natural frequencies of vibration of the castings. One environmental factor which has an effect on the natural frequencies of vibration of a structure, and therefore may have an effect on the success of this method, is ambient temperature. For most applications, the effect of minor changes in temperature on the mechanical performance of materials is negligible. However, this application may require natural frequencies to be measured to a high degree of accuracy. In this case, the small effect of temperature change could be significant.

The natural frequencies of vibration of a structure are a function of the distribution of mass and stiffness of the structure, and in theory, may be determined analytically from the material and geometric properties of the structure. This involves setting up the differential equations governing the motion of the structure, according to the theory of elasticity, and then solving these equations. In practice, for any geometry but the simplest shapes, solution of the equations of motion is an extremely difficult task. In most cases, a closed form solution is not attempted, but numerical approximations such as the finite element method are used instead. This section will attempt to determine the order of magnitude of temperature effects on the natural frequencies of castings by examining how temperature affects the natural frequencies of simple geometrics for which known solutions to the equations of motion exist. Temperature effects on axial vibration of uniform bars, and flexural vibration of uniform beams will be calculated.

3.5.1 Axial Vibration of Uniform Bars

The natural frequencies of vibration of a uniform bar with free and conditions in axial vibration may be calculated from the equation (Reference 22):

$$\omega_n = \frac{n\pi}{L}\left(\frac{E}{\rho}\right)^{1/2} \qquad (1)$$

where: ω_n = circular natural frequency of mode n (rad/sec)
E = elastic modulus
L = length
ρ = mass density
n = mode number

Adams et all (Reference 17) have derived an expression for the relative change in natural frequency due to small variations in the length, elastic modulus and mass density of a uniform bar as follows:

$$\frac{\delta\omega_n}{\omega_n} = -\frac{\delta L}{L} + \frac{1}{2}\frac{\delta E}{E} - \frac{1}{2}\frac{\delta\rho}{\rho} \qquad (2)$$

where δ denotes small changes in each parameters.

By definition, the temperature coefficient of linear expansion is:

$$\alpha_L = \frac{\delta L}{L}\cdot\frac{1}{\delta T} \qquad (3)$$

Since the mass is unaffected by temperature, small changes in the mass density may be expressed as:

$$\delta\rho = \frac{m}{\delta V} \qquad (4)$$

Also, the temperature coefficient of volume expansion is approximately:

$$\alpha_V = 3\,\alpha_L \qquad (5)$$

The temperature coefficient of elastic modulus may be defined in the same manner as the coefficient of linear expansion:

$$\alpha_E = \frac{1}{E} \frac{\delta E}{\delta T} \tag{6}$$

Substituting expressions (3) to (6) into (2) and rearranging yields:

$$\frac{\delta \omega_n}{\omega_n} = \frac{1}{2} \left(\alpha_L + \alpha_E\right) \delta T \tag{7}$$

or

$$\frac{\delta \omega_n}{\delta T} = \frac{1}{2} \left(\alpha_L + \alpha_E\right) \omega_n \tag{8}$$

The Steel Casting Handbook [SCH] (Reference 23) lists values for the coefficient of linear expansion of various steel alloys. A typical value is:

$$\alpha_L = \frac{12\ \mu m}{m°K} \tag{9}$$

The temperature coefficient of modulus for steel alloys may be determined approximately from curves of modulus versus temperature for various alloys (SCH Figure 24-2). The curves are parallel, having a slope of approximately -72 MN/°C. Using a nominal value for elastic modulus of 207 GN/°C, the temperature coefficient of elastic modulus is:

$$\alpha_E = -3.42 \times 10^{-4}\ °C^{-1} \tag{10}$$

Substituting these numerical values into equation (8) we find the effect of temperature of natural frequency of a uniform bar:

$$\frac{\delta \omega_n}{\delta T} = -1.65 \times 10^{-4}\ \omega_n \tag{11}$$

For a natural frequency of 2500 Hz, this corresponds to a frequency shift of -0.41 Hz/°C.

3.5.2 Uniform Beams in Flexural Vibration

The flexural natural frequencies of a uniform beam having a rectangular cross-section of depth h is (Reference 22):

$$\omega_n = k_n \frac{h}{L^2} \left(\frac{E}{\rho}\right)^{1/2} \tag{12}$$

Using an analysis similar to the above, we find the relative effect of a small change in temperature on natural frequency:

$$\frac{\delta\omega_n}{\omega_n} = -2\frac{\delta L}{L} + \delta h + \frac{1}{2}\frac{\delta E}{E} + \frac{1}{2}\frac{\delta\rho}{\rho} \tag{13}$$

But, as above

$$\frac{\delta L}{L} = \alpha_L \delta T \qquad \frac{\delta\rho}{\rho} = -3\alpha_L \delta T \tag{14}$$

and

$$\delta h = h \alpha_L \delta T \tag{15}$$

Substituting (14) and (15) into (13) yields:

$$\frac{\delta\omega_n}{\delta T} = \omega_n \left[\alpha_L \left(h - \frac{1}{2}\right) + \frac{1}{2}\alpha_E\right] \tag{16}$$

Values of temperature coefficients of modulus and linear expansion are as determined above. For example, a uniform beam having a section depth of 50 mm, and a natural frequency of 2500 Hz, equation (16) yields a temperature effect of -0.44 Hz/°C.

These calculations are of course only valid for the simple geometries that they pertain to. To determine if a complex geometric shape experienced temperature effects of the same order of magnitude, a simple test was conducted as follows. A steel casting weighing approximately 19 kg was tested at room temperature to determine its first three natural frequencies of vibration. The casting was then placed in an environment controlled room, and cooled to approximately 10°C below ambient temperature. The three natural frequencies were again measured at this reduced temperature. Finally, the casting was warmed to 10°C above ambient temperature, and the natural frequencies measured again. The results of this simple experiment are shown graphically in Figure 14. For this particular casting the effects of temperature on the three natural frequencies of vibration vary from 0.39 to 0.50 Hz/°C. These values are the same order of magnitude as the calculated temperature effects.

The above calculations and simple experiment indicate that significant natural frequency shifts may be caused by relatively small ambient temperature changes. Thus for the vibration response technique to have consistent results, the temperature must either be controlled, or it must be measured and compensated for. In the experimental work done in the balance of this study, care was taken to ensure that all natural frequency tests were conducted at the same temperature.

3.6 Dimensional Tolerance Effects

The steel casting process is subject to variability which causes the properties of cast components to vary within a range of tolerances. These properties include chemical composition, strength, mass and dimensions.

The property which is most likely to affect the natural frequencies of vibration of a component, is dimension. The geometry of a structure is a measure of its distribution of mass and stiffness, two very important parameters affecting its free vibration. Dimensional tolerances would therefore be very important to a scheme which intends to use natural frequency measurements as a quality control tool. If the variation in natural frequency among castings due to dimensional tolerances is greater than that due to the presence of defects, the method will not work. Therefore, it will be useful to have some indication of how dimensional tolerances affect natural frequencies.

In the steel casting industry, a series of five tolerance grades have been developed, based on the relationship of dimensional tolerances to casting weight and overall dimension. These are listed in the Steel Casting Handbok (Reference 23, Sect. 13-19). The five dimensional tolerance grades cover methods of production which vary from investment, shell, and no-bake molding methods, through to green sand molding processes, where no pattern changes are available.

As noted in Section 3.5, it is theoretically possible to determine the natural frequencies of a casting from its material properties and geometry by solving the differential equations of motion. However, due to the complexity of the equations, this is only practical for the simplest of shapes. To obtain order of magnitude estimates of the effect of dimensional tolerances on the natural frequencies of vibration of castings, the effects of dimensional tolerances on the natural frequencies of uniform bars in axial vibration and flexure were calculated. These simple geometries were used purely for convenience. While the results cannot be applied to more complex geometries, they are useful for the information they provide about the order of magnitude of dimensional tolerance effects.

3.6.1 Axial Vibration of Uniform Bars

The equation for the natural frequencies of axial vibration of a uniform bar was given in section 3.5.

$$\omega_n = \frac{n\pi}{L}\left(\frac{E}{\rho}\right)^{1/2} \tag{1}$$

Consider the effect of changing the length by an amount $\pm T$, while the elastic modulus and mass density are held constant.

$$\omega_{n1} = n\pi\left(\frac{E}{\rho}\right)^{1/2}\frac{1}{(L+T)} \quad ; \quad \omega_{n2} = n\pi\left(\frac{E}{\rho}\right)^{1/2}\frac{1}{(L-T)} \tag{17}$$

The relative change in natural frequency due to this variation in length L is given by

$$\frac{\Delta\omega_n}{\omega_n} = \frac{\omega_{n1}-\omega_{n2}}{\omega_n} = \frac{-2T}{L-(T^2/L)} \tag{18}$$

For practical purposes, $L \gg \frac{T^2}{L}$, and (18) simplifies to

$$\frac{\Delta\omega_n}{\omega_n} = \frac{-2T}{L} \tag{19}$$

3.6.2 Uniform Rectangular Bars in Flexure

The equation for the natural frequencies of vibration of a uniform rectangular bar in flexure was given in section 3.5

$$\omega_n = \frac{k_n h}{L^2}\left(\frac{E}{\rho}\right)^{1/2} \tag{12}$$

Note that both the section depth, h, and the beam length, L, are subject to dimensional tolerances, and variations in these dimensions are not necessarily related to each other. However, the effect of length variation is a second order effect in flexure, whereas section depth is a first order effect. Therefore, in this analysis, the length is assumed to be held constant, along with the elastic modulus and mass density.

The relative change in natural frequency due to a variation of $\pm T$ in dimension h is then

$$\frac{\Delta\omega_n}{\omega_n} = \frac{(h+T) - (h-T)}{h} = \frac{2T}{h} \qquad (20)$$

Hence the effect of depth tolerance on the natural frequencies of a rectangular beam is equal to the effect of length tolerance on the axial vibration of a uniform bar. The signs indicate that an increase in section depth of a beam increases its natural frequency, while an increase in the length of a bar decreases its natural frequency.

The steel casting handbook gives a series of empirical equations to calculate the magnitude of dimensional tolerances in each of the five casting grades. These equations are of the form

$$T_n = aW^{1/3} + bL^{1/3} \qquad (21)$$

where: T = dimensional tolerance for grade n.
a,b = empirically derived constants.
w = casting weight.
L = dimension for which tolerance is required.

Using equation (21) in conjunction with equations (19) and (20), it is possible to predict the relative effect of length and depth tolerances within each grade of casting on the natural frequencies of uniform bars in axial vibration, and uniform beams in flexure. Figure 15[a] is a graph of the relative change in natural frequency versus dimension for several different casting weights in casting tolerance grade T3, the grade with the best tolerances. Figure 15[b] is a similar graph for casting tolerance grade T7, the worst tolerance grade. The following example will help to interpret the graphs. The natural frequency in axial vibration of a uniform bar of length 10 inches and weight 10 lb in casting grade T3 could be expected to vary within 1% of its nominal value due to dimensional tolerances. If its length were 4 inches, the natural frequency could be expected to vary within 2% of its nominal value.

To verify the above calculations, a simple experiment was conducted as follows. Three uniform mild steel bars of length 12.0 inches were machined to diameters of 0.810 inches, 0.890 inches, and 0.970 inches. This corresponds to the tolerance variation in Grade T7. The first two natural frequencies of each of the bars in flexure was measured in the laboratory. Following this, two of the bars were cut to shorter lengths by 0.110 inches and 0.220 inches, respectively, again corresponding to tolerance grade T7. The first natural frequency of the bars in axial vibration was then measured. The measured changes in natural frequencies were then

compared to results calculated as described above. Results of this experiment are given in Table 1. As the results indicate, there is a very good correlation between the measured and calculated changes in natural frequency.

The above analysis is of course a gross simplification. In practice, dimensions will not vary independently of one another nor will they vary uniformly. However, the analysis serves to indicate the order of magnitude of dimensional tolerance effects, and it demonstrates that dimensional tolerance effects are quite significant.

To gain further insight into the importance of dimensional tolerances to the vibration response of castings, a sample of 50 small precision investment castings was obtained for analysis purposes. The selected casting is shown in Figure 16. It is an artificial knee joint used for prosthetic limbs. The casting was made of a steel alloy, weighed about 115 grams, and had extremely good surface finish properties. The batch of 50 pieces was borrowed from the inventory of a local manufacturer of prosthetic devices. There was no reason to suspect that any of the castings was defective. To verify this, all 50 pieces were radiographed, to search for internal defects. No defects were found.

On receiving the castings, the weight and selected dimensions were measured for each piece. Figure 16 shows the dimensions that were arbitrarily selected. The first and second natural frequencies of vibration of each casting were also measured using the impact test method. The frequencies were determined from peaks in the sound pressure level response spectra.

Table 3 lists the measured natural frequencies, weights, and dimensions for all 50 pieces. The means, standard deviations, and coefficients of variation of the measured variables are shown in Table 3. Note that for most of the dimensions, the coefficient of variation is in the order of 0.22% or less, indicating the casting had very good tolerances. The coefficient of variation of weight is slightly higher, 0.32%, while for the natural frequencies it is still higher, at 0.58 to 0.62%.

Figure 17 shows histograms of frequency of occurrence versus natural frequency of vibration for the first and second natural frequencies. Confidence limits for 90% to 99% probability levels are also shown on the histograms. Obviously, the natural frequencies of vibration follow a more or less normal distribution. Note that three out of the 50 castings have natural frequencies which are above the 98% confidence limits. Because the natural frequencies of these three castings were so different from the mean, it was suspected that they may have contained defects that were not detected in the original radiographs. The three were

sent out for further radiographic inspection from different angles (report in Appendix), but were found to be completely free of internal defects.

The above example shows that even for a precision casting with extremely tight tolerances, the castings' natural frequencies of vibration vary within certain limits. This variation in natural frequencies is not caused by defects, but by normal variations in mechanical properties, such as dimensions, which result from the casting process.

This analysis suggests that rather than try to account for the effects of variation in dimensions or other mechanical properties on the natural frequencies of vibration of a casting, statistical techniques should be used. In practice, the vibration response method will probably require that a statistical analysis of a representative number of acceptable castings be conducted, in order to establish a reference standard against which the vibration response of other castings can be compared. By using statistical techniques, the effects of all parameters, including dimensions, which affect the vibration response, would be taken care of automatically.

3.7 Test Method Development Summary

In summary, the following highlights are a result of the test method development phase of this study.

1. Of the three modal properties considered for the vibration response method, natural frequency, damping factor, and mode shape, it appears practical only to measure natural frequencies for the purpose of this study.

2. Impact excitation is suitable for the vibration response method applied to castings. This excitation method uses a simple test setup, and provides reasonably consistent results.

3. Since natural frequency is the only parameter which will be measured, it is not necessary to measure the excitation force. The natural frequency can be determined directly from the vibration response spectrum.

4. It will probably be necessary to use the zoom transform of the FFT analyzer, to achieve sufficient frequency resolution .

5. Temperature changes of a few degrees C cause a measureable shift in the natural frequencies of vibration of steel

castings. The importance of this effect is as yet unknown, but this variable can be removed by holding the test or temperature constant, or by correcting the measured data for temperature changes.

6. Dimensional tolerances of steel castings will have a significant effect on their vibration response, and hence on the application of this method. To account for the effect of dimension and other mechanical properties which undergo normal variations due to the casting process, it is expected that statistical techniques will be required to establish a pass/fail criterion based on natural frequency.

The exact nature of the statistical analysis which would be used to establish a pass/fail criterion based on vibration response tests was unknown at the conclusion of the test method development phase. Before a suitable analysis could be determined, it was necessary to have a representative set of empirical data to work with. The case study presented in the next section represents the application of the vibration response method to a production casting. Standard radiography techniques were used with the vibration response method to gather the required data. Therefore, the data analysis procedure will be presented with the case study, in the next section.

4.0 CASE STUDY

To verify the vibration response test method developed, a typical foundry casting was subjected to a vibration response study. Various defects such as porosity and inclusions were present in varying levels in some of the castings. An attempt was made to establish a correlation between the level of defects and natural frequency of the cast components.

4.1 Sample Description

A chisel plow tooth casting (Figure 19) used for zero tillage, was supplied by the foundry, Abex Industries Ltd. of Selkirk, Manitoba. It weighed approximately 1042 grams and was a quenched/tempered low alloy steel cast by a shell molding process. A total of 81 castings were received in three separate groups. One group contained sand inclusions deliberately added to them. Each casting group was made at different heats.

4.2 Procedure

All castings were weighed, measured for specific dimensions (Table 4 and Table 5), and radiographed before testing for vibration response. Radiographs provided by the foundry for each casting were evaluated in terms of shrinkage, porosity and inclusions.

The plow tooth castings were then tested using a small impact hammer (PCB Model 086 B03, Signal Conditioner Model 480 D06) and a sound level meter (Bruel & Kjaer Model 2230) as a transducer to measure the vibration response. The signal served as the input to a Zonic 6080 Multichannel FFT Analyzer for analysis. The first two natural frequencies were noted from the frequency spectrum obtained as an average of three impacts and recorded for each casting.

4.3 Results

Depending upon the severity of the defects in the castings, radiographs were assigned a level 1 to 4. Positive prints of typical castings in these levels are shown in Figure 20 (a-d). These figures show a level 1 casting with only minor shrinkage near the gate and a level 4 casting with excessive porosity/inclusions. These levels are assigned subjectively and do not conform to any industry standard.

The first two natural frequencies of all chisel plow tooth castings are shown in Tables 5 and 6 which also list the weight of each casting and its selected dimensions. The probability of occurrence of the first and second natural frequency for these castings is shown as a histogram in Figure 21.

4.4 Discussion

A review of Figure 21 indicates that the natural frequency of the castings fall into two or three major groups. To relate the frequency level of the first natural frequency of plow teeth, each casting within a frequency group in Figure 21 was assigned its own radiographic level and an "average" value was determined. This average value is marked on the top of each bar in the histogram of Figure 22. The same data can be presented as a plot of first natural frequency and an average radiographic level of castings as shown in Figure 23. This figure indicates that as the quality of castings improves, so does their natural frequency.

It is, therefore, possible to accurately determine the structural integrity of the cast product using the vibration response method and an initial calibration exercise. This calibration exercise would

consist of measuring the vibration response of known quality castings to determine the correlation with frequency response and to establish an acceptance limit based on frequency. In the case study presented, it may be deemed by the manufacturer that a quality level of 2 is the minimum requirement. The calibration exercise would indicate that a first frequency of 5645 Hz is the lower acceptable frequency limit (Figure 23). Then in all subsequent production, only a vibration response measurement would be required to determine whether a casting is acceptable or not.

5. CONCLUSIONS

The test method development and the application of the method in the case study permit the following conclusions to be made:

1. The vibration response method of determining the presence of defects in steel cast components is feasible.

2. The optimum method utilizes impact for excitation and a sound level meter (microphone) to measure the response. This combination proved to be the most practical given the limitations of other transducers and the environment in which the transducers would be used.

3. As demonstrated in the case study, the application of the method is straight forward, requiring only a calibration exercise and simple subsequent measurement which are compared to a lower acceptable level.

REFERENCES

1. Emerson, P. J. and Simmons, W., "Final Report on the Evaluation of the Graphite Form in Ferritic Ductile Iron by Ultrasonic and Sonic Testing, and the Effect of Graphite Form on Mechanical Properties" AFS Transactions, Vol.84 p.109-128 (1976).

2. Fuffer, A. G., "Evaluation of the Graphite Form in Pearlitic Ductile Iron by Ultrasonic and sonic Testing, and the Effect of Graphite Form on Mechanical Properties", AFS Trans., Vol.85, p.509-526 (1977).

3. Fuller, A. G. "Effect of Graphite Form on Fatigue Properties of Pearlitic Ductile Irons", AFS Tran. vol.85 p.527-536 (1977).

4. Fuller, A. G. Emerson, P. J., and Sergent, G. F. "Report Upon the Effect Upon Mechanical Properties of Variation in Graphite Form in Irons Having Various Amounts of Ferrite and Pearlite in the Matrix Structure, and the Use of Nondestructive Tests in the Assessment of Mechanical Properties of Such Irons", AFS Trans. Vol.88, p.21-50 (1980).

5. Fuller, A. G. "Nondestructive Assessment of the Properties of Ductile Iron Castings", AFS Trans. Vol.88, p.751-768 (1980).

6. Rhoda, D.A., "Sonic Testing of Gray Iron Sleeve Castings to Predict Tensile Strength", SAE Earthmoving Industry Conference, Peoria, Illinois, #690371, April 1969.

7. Kovacs, B. V., "Quality Control and Assurance by Sonic Resonance in Ductile Iron Castings" AFS Trans, Vol. 84, p. 499-508 (1976).

8. Adams, R. D., and Vaughan, N.D., "Resonance Testing of Cast Iron", Jour. of Non-descructive Evaluation, Vol. 2 (1) p. 65-74 (1981).

9. Breeze, J. E. and Nilberg, R. H., "Predicting by Sonic Measurements the Strength of Logs and Poles Having Internal Decay", Forest Products Research, Vol. 21, p. 39-43 (1971).

10. Bodig, J., Goodman, J. R., Phillips, G. K. and Landers, P.G. "Reliability of Wood Transmission Structures through NDE", IEEE Trans., Power Apparatus and Systems, Vol. PAS - 101, #8, p. 2409-24, (1982).

11. Bluvshtein, M.N. et al "Sonic Method for Checking the Thermal Shock Resistance of Steel Casting Plugs." Translated from Ogneupory, No. 5, p. 4 - 10 (1972).

12. Alexander, A.M., "Development of Procedures for Nondestructive Testing of Concrete Structures, Report 2., Feasibility of Sonic Pulse - Echo Technique" Report No. C-77-11, Office, Chief of Engineers, U. S. Army, Washington, D.C. April (1980).

13. Renekar, D. H. et al, "An NDT Method Using Piezoelectric Polymer Transducers and Computerized Vibration Spectrocscopy", NDT International, p. 15 -16, February (1978).

14. Eshleman, R. L., "Machinery Diagnostic and Your FFT", Sound and Vibration, p. 12 - 18, April (1983).

15. Baird, B. "Vibration Detection of Component Operability" ASME paper 76-ENAS-18 (1978).

16. Smiley, R. G. Structural Dynamics Research Corp., Cincinnati, Private Communications.

17. Adams R.D. et al. "A Vibration Technique for Nondestructively Assessing the Integrity of Structures" J. Mech. Eng. Sci., 2 (20) p. 93 - 100 (1978).

18. Harris, W. J. "Report of a Preliminary Survey of NDT Technology with Special Reference to Offshore Structures", British J. of NDT, p. 117 - 120, May 1983.

19. Upchurch, B. L., Furgason, E.S., and Miles, G.E. "Spectral Analysis of Acoustical Signal for Damage Detection" Paper No.85-6014 American Society of Agricultural Engineers Meeting June 23-26 (1985).

20. Pye, C.J. and Adams R. D. "Detection of Damage in Fibre Reinforced Plastics using Thermal Fields Generated during Resonant Vibration" NDT International, p. 111 - 118, June 1981.

21. "Shock and Vibration Handbook" - Ed. G.J. Harris and C.E. Crede, McGraw Hill 1961.

22. Craig, Ray R. Jr., "Structural Dynamics" - An Introduction to Computer Methods, John Wiley and Sons, 1981.

23. "Steel Casting Handbook" Ed. P.F. Wieser, Steel Founders Society of America, 1980.

24. Ewins, D. J., "Modal Testing - Theory and Practice", John Wiley and Sons

Table 1

EFFECT OF DIMENSIONAL CHANGES ON NATURAL FREQUENCY

Shift in natural frequencies of vibration of machined bars due to changes in length and diameter corresponding to casting tolerance grade T7.

Vibration Mode	Change in Natural Frequency (%)	
	Predicted	Measured
First axial	1.9	1.9
First bending	17.6	17.2
Second bending	17.6	16.4

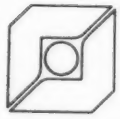


Table 2

SAMPLE: ARTIFICIAL KNEE JOINT INVESTMENT CASTING

MEASURED DATA

Sample Number	First Freq.	Second Freq.	Weight Grams	Dim. a inches	Dim. b inches	Dim. c inches	Dim. d inches	Dim. e inches
1	3855	5891	114.43	1.744	0.360	0.358	0.947	0.946
2	3852	5873	114.62	1.747	0.361	0.359	0.947	0.946
3	3862	5896	114.54	1.745	0.360	0.358	0.945	0.946
4	3903	5940	115.06	1.747	0.361	0.359	0.947	0.944
5	3855	5875	114.14	1.744	0.360	0.358	0.947	0.946
6	3846	5859	113.97	1.743	0.359	0.357	0.946	0.946
7	3854	5878	114.23	1.746	0.360	0.361	0.947	0.947
8	3852	5887	114.38	1.747	0.360	0.360	0.946	0.945
9	3849	5861	114.23	1.748	0.360	0.359	0.946	0.947
10	3853	5898	113.99	1.743	0.360	0.359	0.944	0.944
11	3867	5902	114.66	1.740	0.361	0.359	0.946	0.947
12	3858	5854	113.77	1.740	0.360	0.358	0.946	0.945
13	3895	5932	114.66	1.738	0.360	0.358	0.946	0.945
14	3856	5873	113.97	1.741	0.361	0.359	0.946	0.946
15	3859	5899	114.13	1.743	0.360	0.358	0.946	0.944
16	3875	5901	114.28	1.744	0.361	0.360	0.945	0.943
17	3843	5881	114.32	1.746	0.361	0.359	0.948	0.947
18	3853	5899	114.74	1.746	0.361	0.359	0.947	0.948
19	3867	5912	114.31	1.747	0.360	0.359	0.946	0.947
20	3844	5864	114.31	1.747	0.360	0.358	0.946	0.944
21	3846	5884	113.94	1.745	0.360	0.359	0.946	0.946
22	3892	5925	114.90	1.747	0.361	0.359	0.946	0.947
23	3856	5863	114.72	1.745	0.362	0.360	0.947	0.945
24	3872	5897	114.45	1.748	0.361	0.360	0.947	0.945
25	3882	5918	114.54	1.741	0.360	0.358	0.946	0.947
26	3835	5862	113.98	1.743	0.359	0.358	0.945	0.945
27	3884	5915	115.15	1.746	0.361	0.360	0.947	0.946

Table 2 (cont'd)

SAMPLE: ARTIFICIAL KNEE JOINT INVESTMENT CASTING

Sample Number	First Freq.	Second Freq.	Weight Grams	Dim. a inches	Dim. b inches	Dim. c inches	Dim. d inches	Dim. e inches
28	3885	5925	114.90	1.743	0.361	0.358	0.945	0.945
29	3834	5857	113.92	1.747	0.360	0.359	0.946	0.944
30	3875	5925	114.88	1.749	0.360	0.360	0.947	0.944
31	3925	5997	114.68	1.749	0.362	0.359	0.946	0.945
32	3820	5836	114.43	1.746	0.359	0.359	0.945	0.944
33	3851	5876	113.84	1.743	0.360	0.359	0.947	0.947
34	3850	5885	114.30	1.742	0.360	0.359	0.946	0.946
35	3934	5983	114.68	1.746	0.363	0.359	0.946	0.946
36	3850	5884	114.05	1.745	0.360	0.359	0.947	0.945
37	3847	5872	114.91	1.747	0.361	0.359	0.948	0.946
38	3922	5976	114.92	1.751	0.362	0.360	0.945	0.946
39	3874	5895	113.99	1.745	0.360	0.359	0.946	0.946
40	3903	5938	114.98	1.740	0.360	0.359	0.947	0.945
41	3857	5882	114.88	1.749	0.361	0.359	0.946	0.948
42	3865	5902	114.57	1.745	0.360	0.360	0.946	0.946
43	3880	5916	114.69	1.742	0.361	0.359	0.947	0.945
44	3825	5818	114.17	1.743	0.360	0.359	0.948	0.946
45	3864	5891	114.90	1.749	0.362	0.360	0.948	0.947
46	3843	5871	114.42	1.744	0.361	0.360	0.949	0.948
47	3853	5897	114.73	1.743	0.360	0.359	0.949	0.947
48	3868	5912	114.19	1.749	0.361	0.360	0.947	0.946
49	3839	5851	113.89	1.746	0.360	0.360	0.947	0.946
50	3855	5874	114.20	1.743	0.360	0.360	0.947	0.945

Table 3

SAMPLE: ARTIFICIAL KNEE JOINT INVESTMENT CASTING

Calculated Data

	First Freq.	Second Freq.	Weight Grams	Dim. a inches	Dim. b inches	Dim. c inches	Dim. d inches	Dim. e inches
Minimum	3820	5818	113.77	1.738	0.359	0.357	0.944	0.943
Maximum	3934	5997	115.15	1.751	0.363	0.361	0.949	0.948
Arith Mean	3864	5894	114.41	1.745	0.360	0.359	0.946	0.946
Std. Dev.	23.96	34.38	0.3670	0.0028	0.0008	0.0008	0.0010	0.0012
CV	0.620	0.583	0.3210	0.159	0.224	0.219	0.106	0.123

Table 4

RADIOGRAPHIC EVALUATION

Rating Level	Description
1	Some shrinkage near gate
2	Level 1 and some gas inclusions
3	Level 1 and moderate gas inclusions
4	Level 1 and excessive gas inclusions

SAMPLE	CASTING	RATING	SAMPLE	CASTING	RATING	SAMPLE	CASTING	RATING
1	1	3	28	16	1	55	19	2
2	2	4	29	6	1	56	15	2
3	3	4	30	5	1	57	36	1
4	4	4	31	18	1	58	15	2
5	5	3	32	11	1	59	24	2
6	6	4	33	4	1	60	14	1
7	7	4	34	8	1	61	21	1
8	8	4	35	9	1	62	18	2
9	9	3	36	10	1	63	19	1
10	10	3	37	1	1	64	17	3
11	11	3	38	3	3	65	12	2
12	12	4	39	21	1	66	16	2
13	13	3	40	34	2	67	4	2
14	14	3	41	29	2	68	9	1
15	15	4	42	24	1	69	22	2
16	16	4	43	31	1	70	13	2
17	17	3	44	27	1	71	6	2
18	18	3	45	28	1	71	10	2
19	19	2	46	32	1	73	2	1
20	20	3	47	35	1	74	11	2
21	21	2	48	22	1	75	23	3
22	2	2	49	20	2	76	8	2
23	7	1	50	30	1	77	1	2
24	14	2	51	23	1	78	7	2
25	12	2	52	26	1	79	3	3
26	13	2	53	33	1	80	5	3
27	25	1	54	17	1	81	20	2

Rating Level	Freq of Occur (1-81)	Freq of Occur (1-21)	Freq of Occur (22-57)	Freq of Occur (58-81)
1	31	0	26	5
2	26	2	9	15
3	15	10	1	4
4	9	9	0	0

TABLE 5

CHISEL PLOW TOOTH CASTING

Measured Data

Sample Number	First Freq.	Second Freq.	Weight Grams	Dim. a inches	Dim. b inches	Dim. c inches	Dim. d inches	Dim. e inches	Dim. f inches
1	5543	8550	1036.5	0.767	0.994	0.273	0.263	5.17	0.15
2	5560	8597	1037.4	0.785	0.986	0.269	0.252	5.17	0.15
3	5532	8535	1044.0	0.776	0.997	0.285	0.267	5.19	0.15
4	5495	8471	1047.6	0.793	0.996	0.277	0.265	5.17	0.14
5	5499	8482	1022.6	0.765	0.989	0.272	0.258	5.15	0.14
6	5435	8394	1029.3	0.777	0.987	0.269	0.265	5.20	0.17
7	5486	8514	1017.7	0.757	0.985	0.261	0.269	5.16	0.16
8	5527	8534	1034.2	0.769	0.996	0.276	0.265	5.15	0.15
9	5550	8608	1048.7	0.788	0.989	0.276	0.262	5.16	0.15
10	5493	8497	1032.0	0.764	0.991	0.275	0.262	5.18	0.16
11	5552	8609	1041.4	0.772	0.998	0.283	0.269	5.19	0.12
12	5552	8592	1031.4	0.776	0.985	0.278	0.253	5.15	0.13
13	5491	8527	1045.1	0.778	0.996	0.281	0.262	5.15	0.16
14	5557	8599	1026.3	0.760	0.986	0.263	0.263	5.15	0.16
15	5482	8469	1028.9	0.770	0.993	0.265	0.259	5.15	0.14
16	5519	8556	1024.8	0.761	0.987	0.266	0.251	5.17	0.18
17	5530	8575	1042.9	0.770	0.996	0.280	0.262	5.17	0.16
18	5554	8649	1034.1	0.759	0.991	0.275	0.261	5.17	0.10
19	5601	8720	1046.7	0.769	0.997	0.278	0.267	5.15	0.11
20	5602	8531	1034.0	0.767	0.997	0.267	0.262	5.15	0.14
21	5497	8499	1025.0	0.767	0.989	0.263	0.264	5.19	0.15
22	5735	8932	1079.3	0.760	1.039	0.303	0.263	5.15	0.13
23	5945	9187	1124.1	0.813	1.047	0.297	0.273	5.13	0.14
24	5711	8891	1070.6	0.777	1.020	0.272	0.272	5.17	0.12
25	5786	8926	1100.5	0.799	1.036	0.270	0.293	5.17	0.12
26	5646	8818	1058.6	0.775	1.009	0.277	0.266	5.15	0.08
27	5759	8947	1085.3	0.783	1.022	0.285	0.268	5.17	0.15

Table 5 (cont'd)

Sample Number	First Freq.	Second Freq.	Weight Grams	Dim. a inches	Dim. b inches	Dim. c inches	Dim. d inches	Dim. e inches	Dim. f inches
28	5800	8961	1113.0	0.815	1.036	0.268	0.293	5.17	0.13
29	5815	8990	1102.6	0.809	1.026	0.281	0.273	5.16	0.16
30	5719	8850	1087.5	0.792	1.030	0.284	0.275	5.16	0.15
31	5752	8935	1079.1	0.782	1.015	0.277	0.269	5.16	0.15
32	5787	8980	1097.8	0.792	1.007	0.293	0.267	5.15	0.18
33	5752	8941	1097.2	0.778	1.014	0.277	0.273	5.17	0.15
34	5640	8776	1054.5	0.763	1.012	0.288	0.254	5.15	0.15
35	5872	9037	1122.7	0.813	1.040	0.283	0.287	5.14	0.18
36	5760	8932	1092.1	0.796	1.035	0.285	0.276	5.16	0.12
37	5864	9098	1115.2	0.790	1.043	0.292	0.275	5.17	0.11
38	5587	8631	1038.5	0.767	1.012	0.278	0.272	5.16	0.15
39	5652	8773	1074.3	0.784	1.005	0.272	0.264	5.17	0.16
40	5756	8994	1084.7	0.777	1.025	0.295	0.265	5.17	0.02
41	5708	8836	1094.8	0.812	1.036	0.277	0.271	5.17	0.15
42	5734	8919	1076.7	0.761	1.024	0.295	0.267	5.14	0.12
43	5760	8918	1090.2	0.783	1.020	0.286	0.265	5.18	0.15
44	5697	8866	1076.7	0.796	1.016	0.269	0.280	5.15	0.14
45	5744	8879	1088.4	0.799	1.021	0.281	0.268	5.16	0.15
46	5923	9181	1114.5	0.804	1.045	0.270	0.295	5.15	0.15
47	5695	8834	1078.7	0.770	1.026	0.288	0.248	5.17	0.20
48	5767	8903	1090.2	0.789	1.034	0.273	0.284	5.17	0.20
49	5644	8829	1066.2	0.764	1.015	0.290	0.267	5.14	0.13
50	5695	8902	1078.1	0.800	1.030	0.268	0.281	5.13	0.16
51	5794	8971	1106.5	0.795	1.028	0.287	0.275	5.15	0.15
52	5815	9004	1100.7	0.789	1.029	0.286	0.275	5.14	0.18
53	5724	8854	1064.9	0.772	1.010	0.273	0.263	5.15	0.13
54	5795	8965	1102.2	0.801	1.022	0.276	0.272	5.16	0.15
55	5553	8752	1053.5	0.782	1.000	0.273	0.268	5.17	0.04
56	5788	8961	1105.5	0.798	1.030	0.290	0.273	5.16	0.18
57	5692	8806	1070.9	0.785	1.018	0.271	0.264	5.15	0.13

Table 5 (cont'd)

Sample Number	First Freq.	Second Freq.	Weight Grams	Dim. a inches	Dim. b inches	Dim. c inches	Dim. d inches	Dim. e inches	Dim. f inches
58	5774	9000	1112.8	0.796	1.038	0.293	0.301	5.16	0.10
59	5655	8839	1108.0	0.792	1.042	0.318	0.275	5.18	0.11
60	5685	8876	1095.6	0.798	1.019	0.284	0.279	5.19	0.12
61	5692	8919	1107.4	0.781	1.048	0.300	0.282	5.18	0.08
62	5709	8919	1101.3	0.780	1.032	0.314	0.284	5.19	0.07
63	5667	8867	1093.7	0.800	1.024	0.299	0.284	5.19	0.06
64	5629	8821	1096.9	0.802	1.039	0.297	0.287	5.18	0.08
65	5658	8875	1096.5	0.782	1.025	0.300	0.278	5.14	0.10
66	5703	8924	1083.0	0.766	1.026	0.296	0.283	5.15	0.07
67	5693	8899	1099.3	0.790	1.035	0.310	0.284	5.15	0.12
68	5701	8922	1090.8	0.769	1.037	0.286	0.304	5.14	0.07
69	5642	8806	1090.6	0.767	1.039	0.305	0.281	5.24	0.12
70	5696	8912	1092.3	0.781	1.018	0.296	0.278	5.18	0.10
71	5786	8970	1118.1	0.808	1.045	0.300	0.295	5.20	0.09
71	5734	8957	1106.6	0.786	1.035	0.279	0.298	5.18	0.11
73	5688	8904	1095.1	0.777	1.034	0.298	0.285	5.16	0.11
74	5726	8892	1100.2	0.776	1.025	0.293	0.281	5.15	0.14
75	5675	8828	1118.5	0.809	1.051	0.297	0.284	5.17	0.10
76	5707	8953	1089.1	0.780	1.023	0.298	0.283	5.17	0.05
77	5722	8907	1098.8	0.790	1.030	0.294	0.288	5.17	0.12
78	5749	8959	1088.8	0.789	1.018	0.292	0.274	5.17	0.13
79	5594	8700	1099.4	0.779	1.032	0.298	0.274	5.18	0.11
80	5575	8771	1084.3	0.775	1.024	0.283	0.293	5.17	0.11
81	5751	8846	1111.2	0.788	1.005	0.308	0.284	5.17	0.19

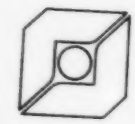
Sample No. Range	Designation
1 - 21	Original batch of castings, no process specified
22 - 57	Control, full HT, Q & T
58 - 81	Sand added, annealed

TABLE 6

SAMPLE: ARTIFICIAL KNEE JOINT INVESTMENT CASTING

Calculated Data

	First Freq.	Second Freq.	Weight Grams	Dim. a inches	Dim. b inches	Dim. c inches	Dim. d inches	Dim. e inches	Dim. f inches
Minimum	5435	8394	1017.7	0.757	0.985	0.261	0.248	5.13	0.02
Maximum	5945	9187	1124.1	0.815	1.051	0.318	0.304	5.24	0.20
Arith Mean	5673	8813	1076.9	0.783	1.018	0.284	0.273	5.16	0.13
Std. Dev.	110.4	180.6	29.47	0.015	0.019	0.013	0.012	0.018	0.038
CV	1.95	2.05	2.74	1.90	1.83	4.49	4.35	0.34	29.75



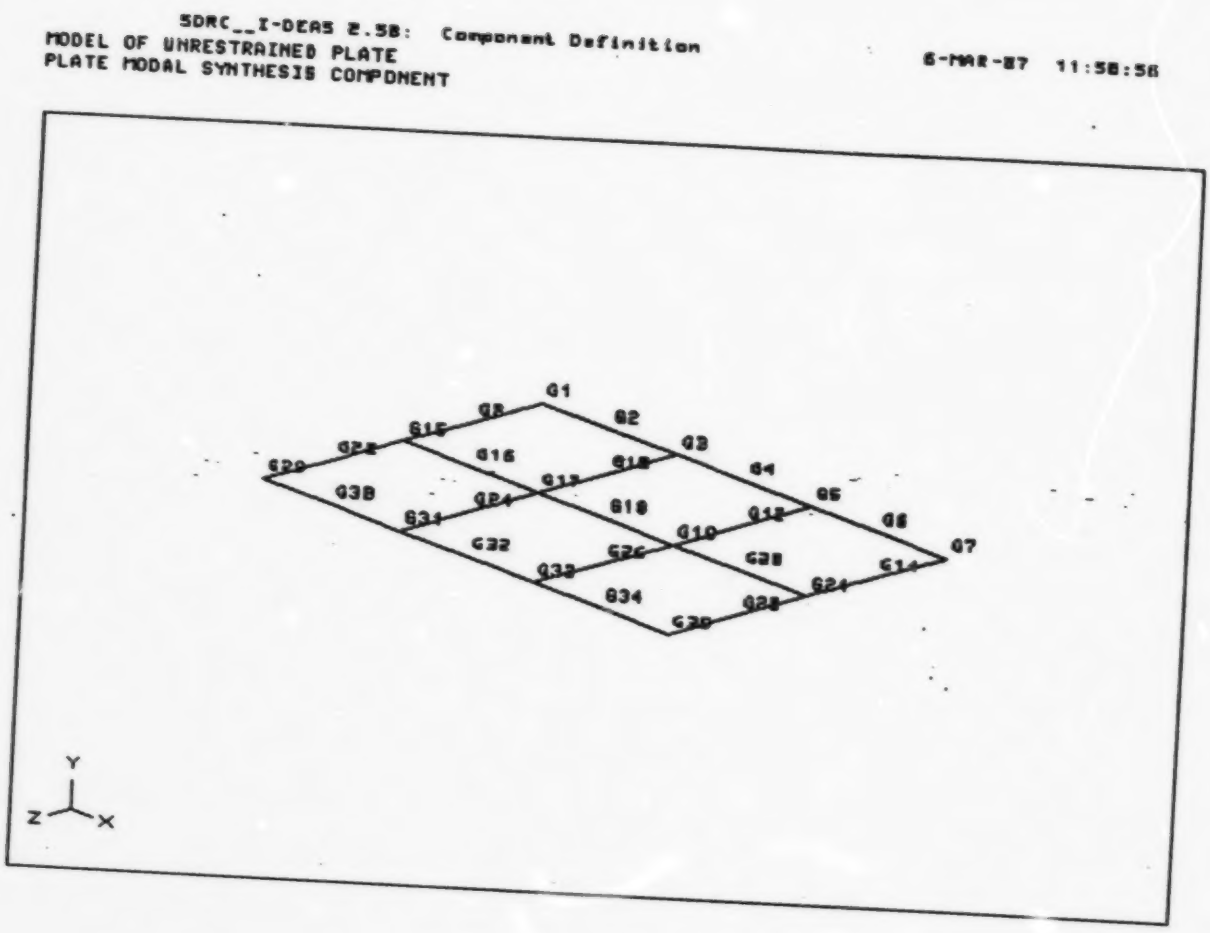


Figure 1. Graphic representation of a computer model of a rectangular steel plate.

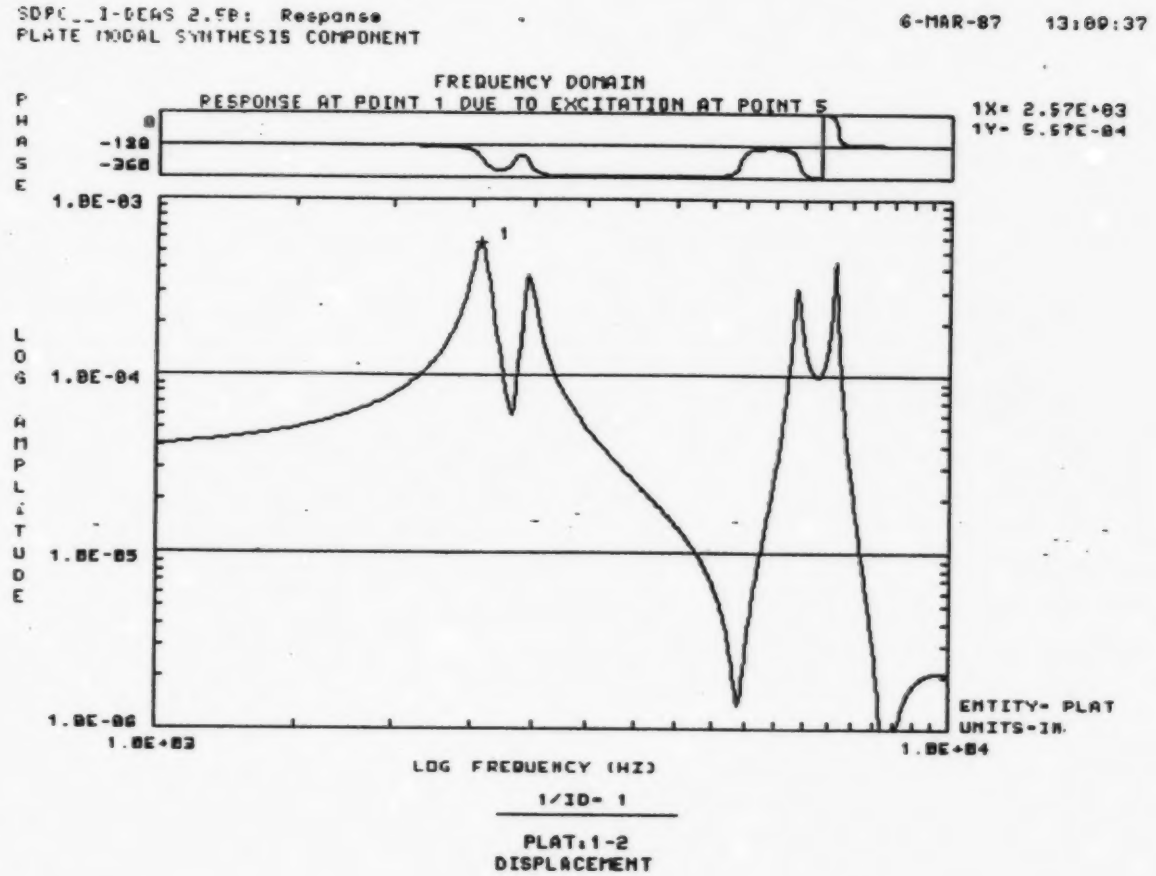


Figure 2. Frequency response function of displacement at G1 on plate due to force applied at G5.

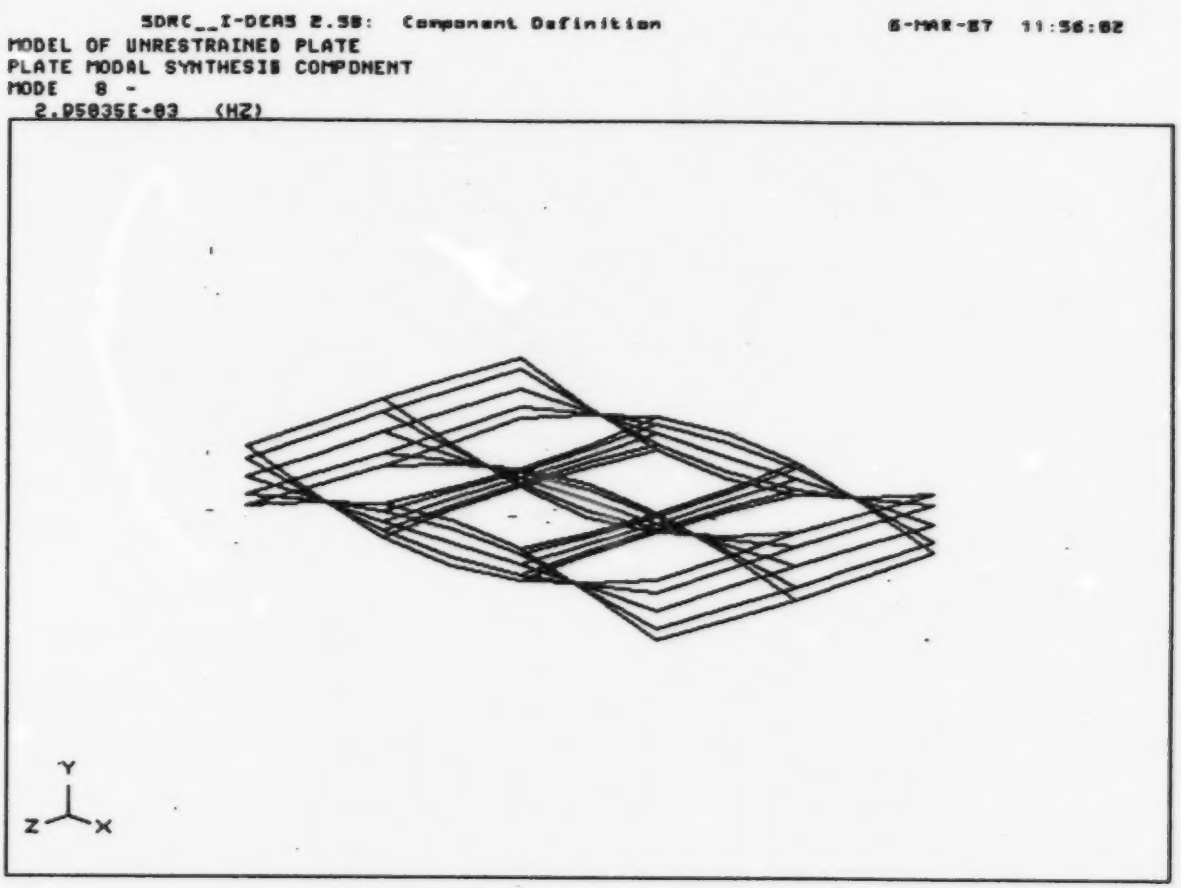


Figure 3. Deformed geometry plot of mode shape of second mode of vibration of steel plate.

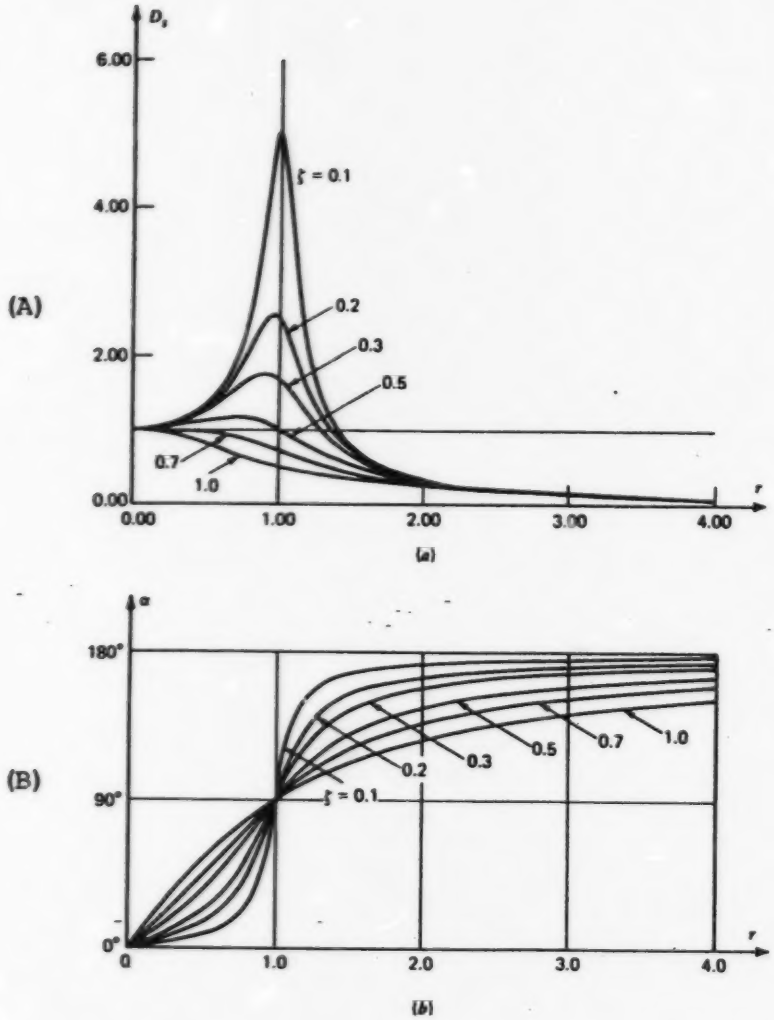


Figure 4. Plot of FRF modulus (a) and phase (b) for various values of the modal damping factor.

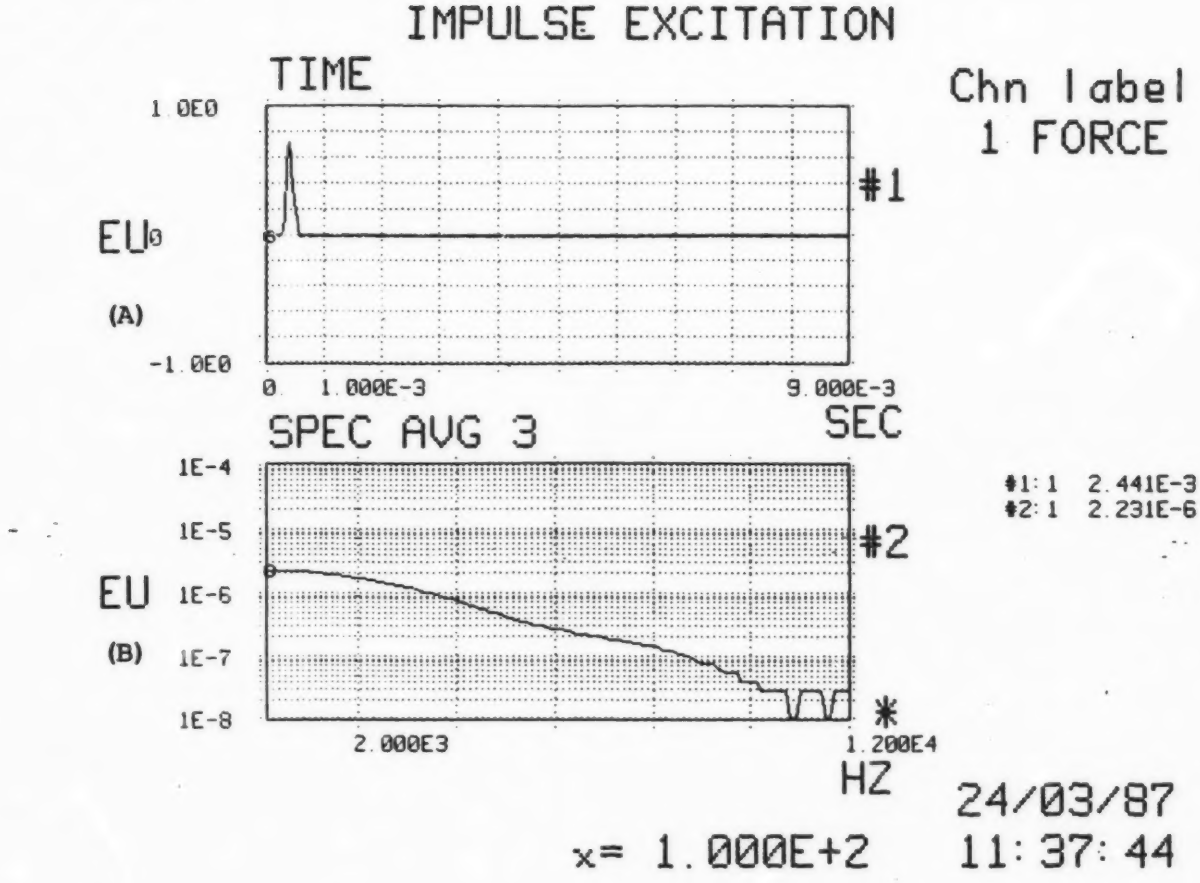


Figure 5. (a) Graph of force versus time for typical impact excitation.
(b) Energy spectral density (spectrum) of the impact force.

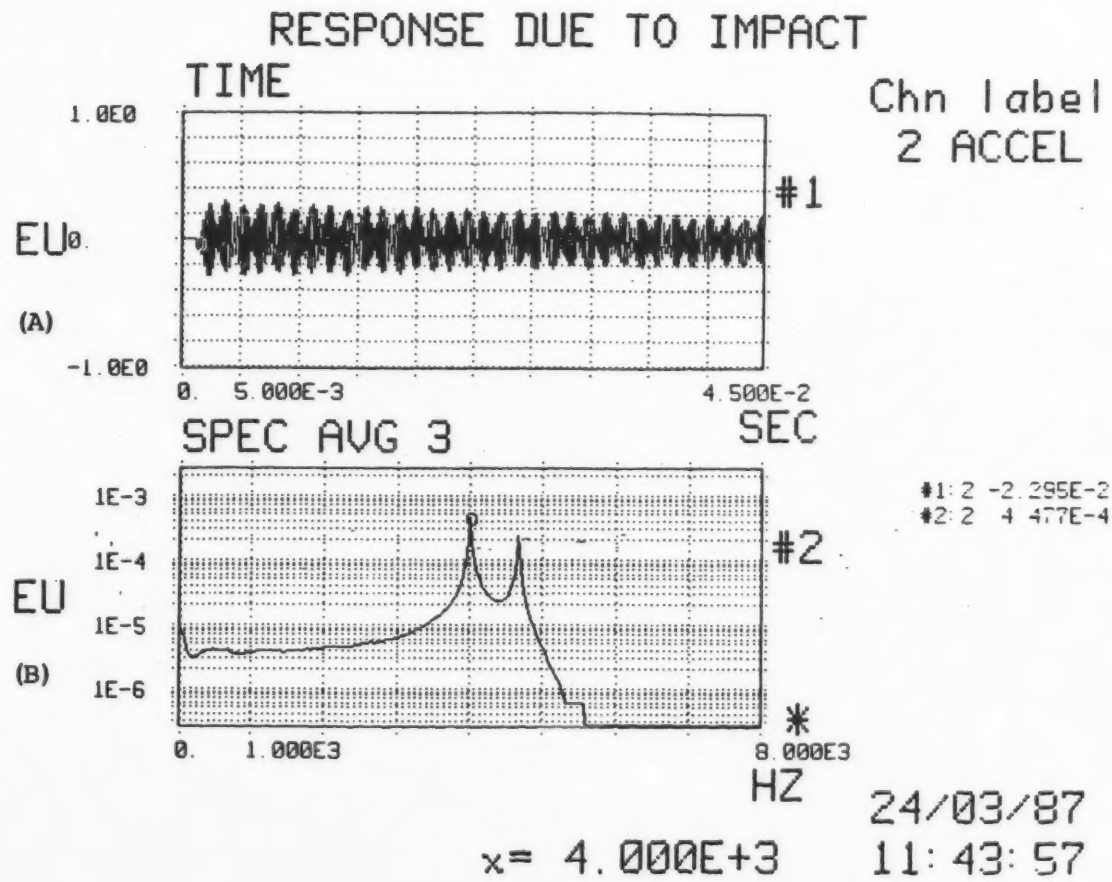


Figure 6. (a) Graph of acceleration versus time for typical response on casting due to impact.
(b) Energy spectral density (spectrum) of the acceleration response.

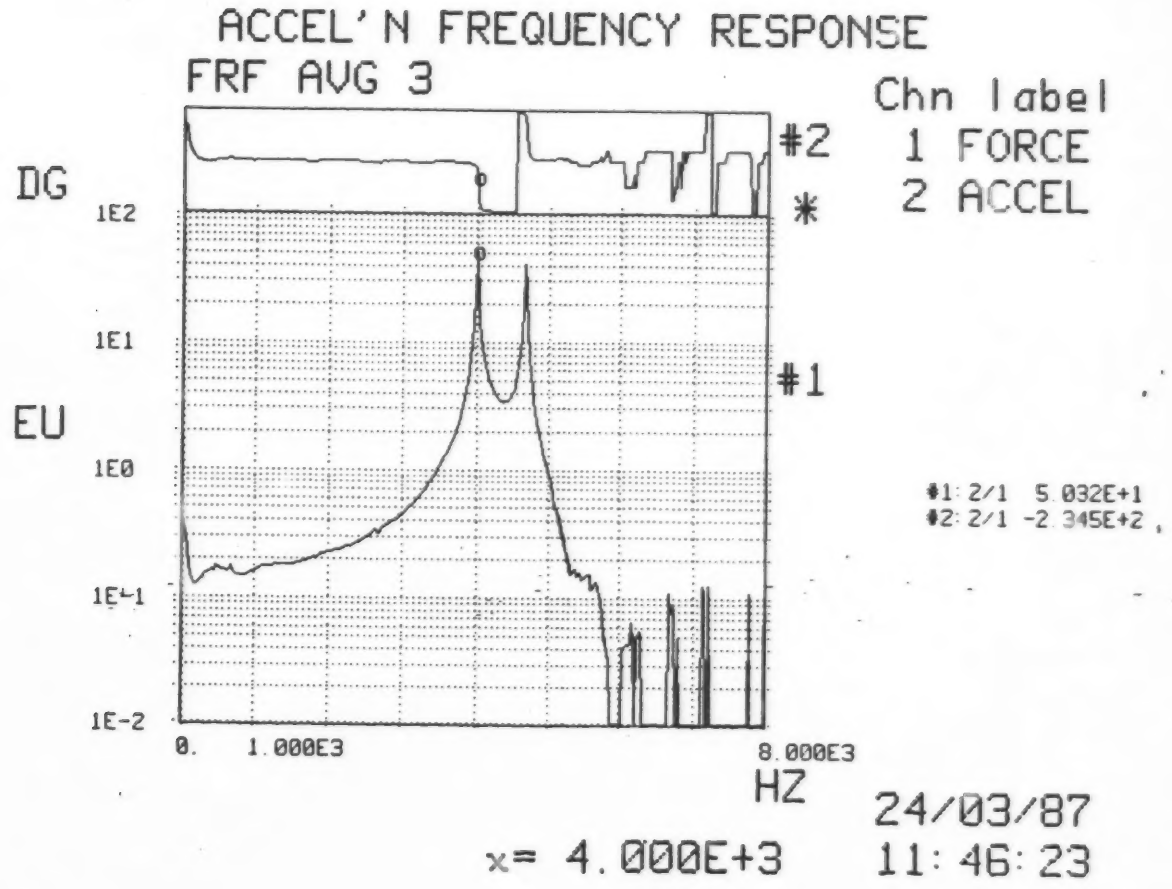


Figure 7. Typical FRF measured on a casting using the impact test method.

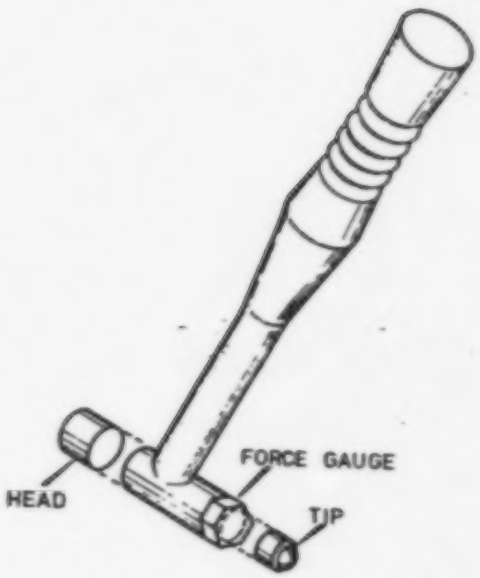


Figure 8. Typical impact test hammer.

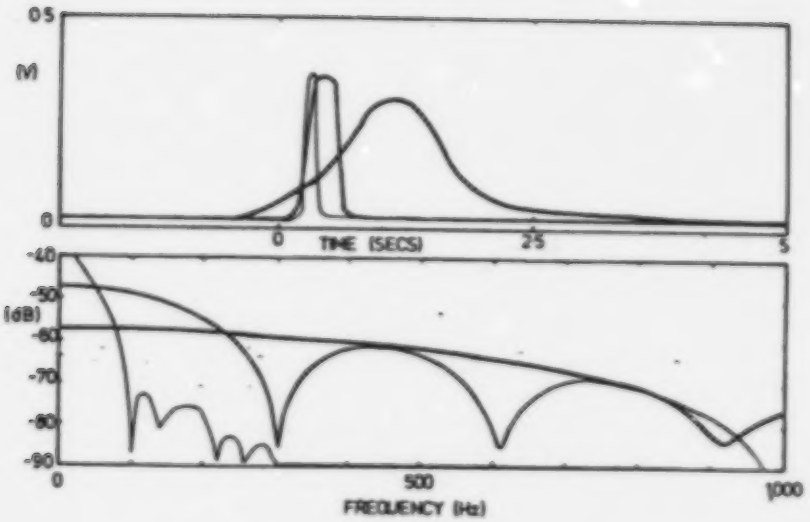


Figure 9. Graph of force versus time for three different impacts (upper curve) and the Fourier transforms of the same impact forces.

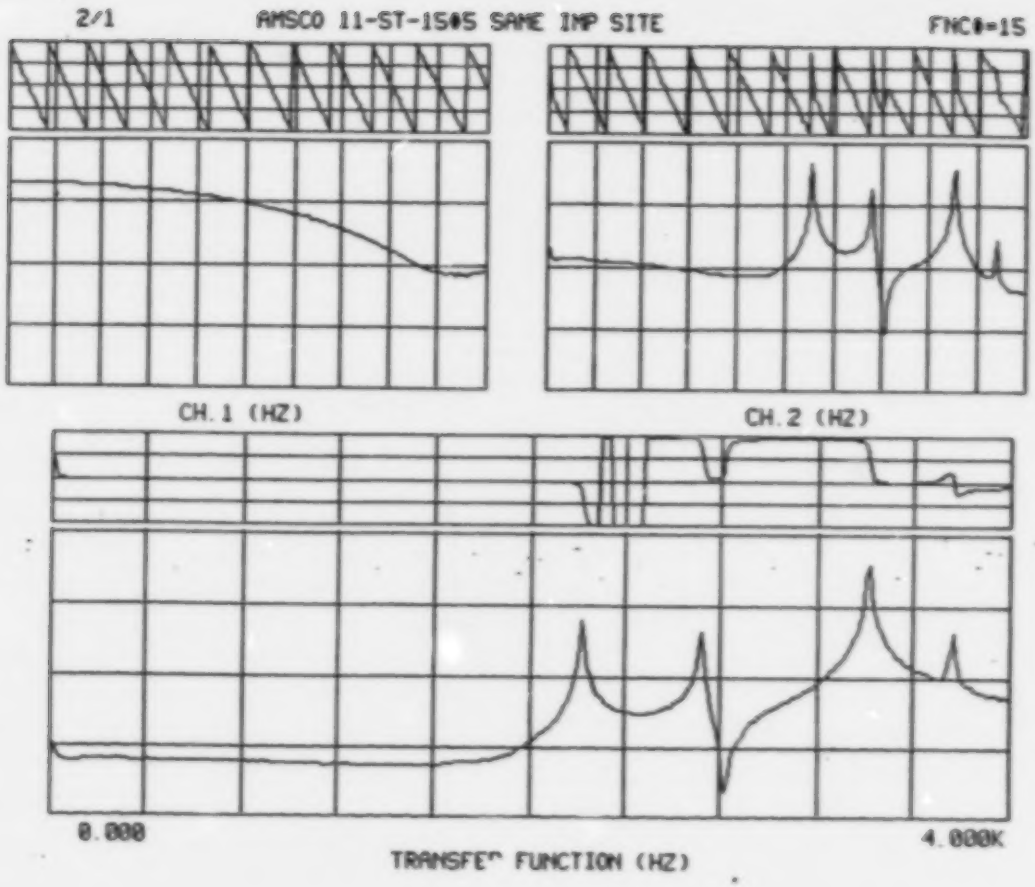


Figure 10. Impact test of a steel casting with baseband analysis frequency of 4.0 kHz; impact force spectrum (upper left), acceleration response spectrum (lower right), and FRF (bottom).

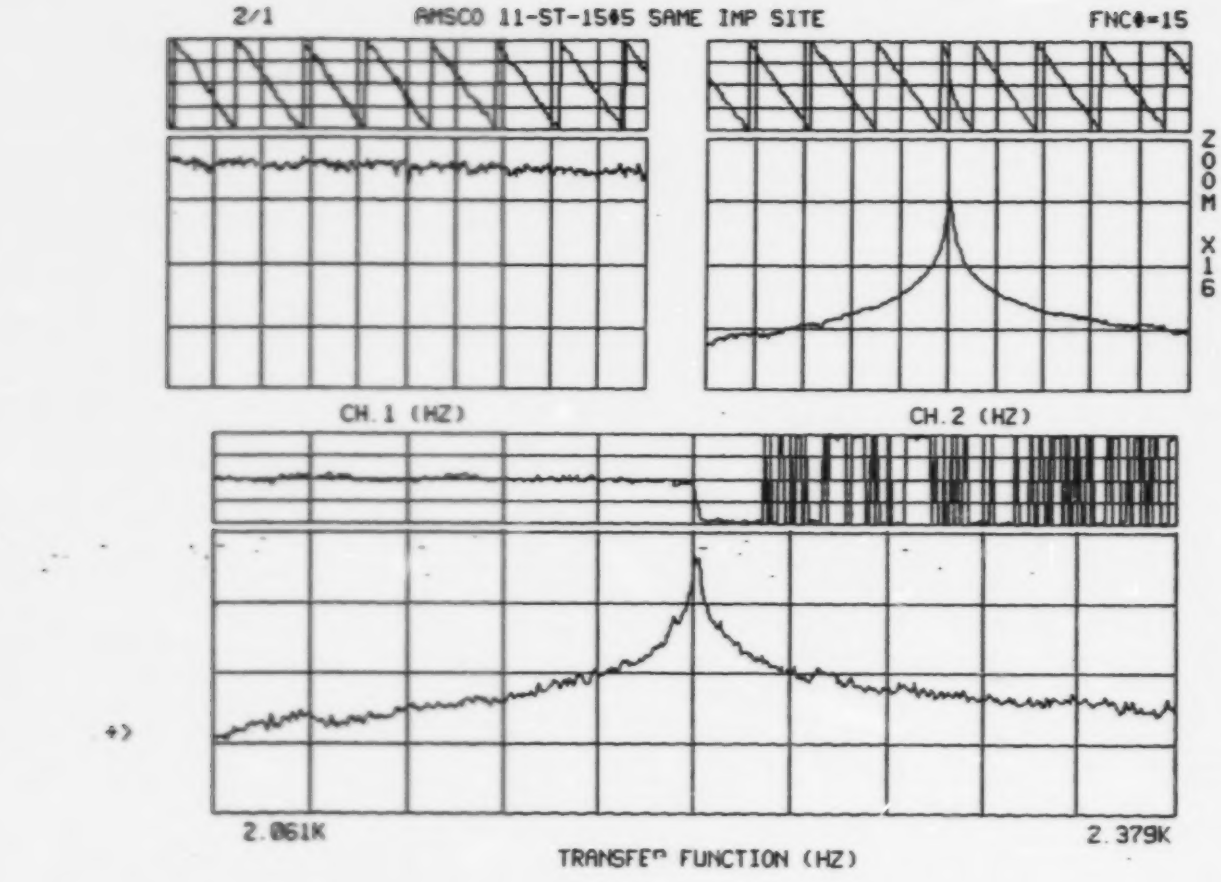


Figure 11. Impact test of a steel casting with baseband analysis frequency of 4.0 kHz and zoom factor of 16; impact force spectrum (upper left), acceleration response spectrum (lower right), and FRF (bottom).

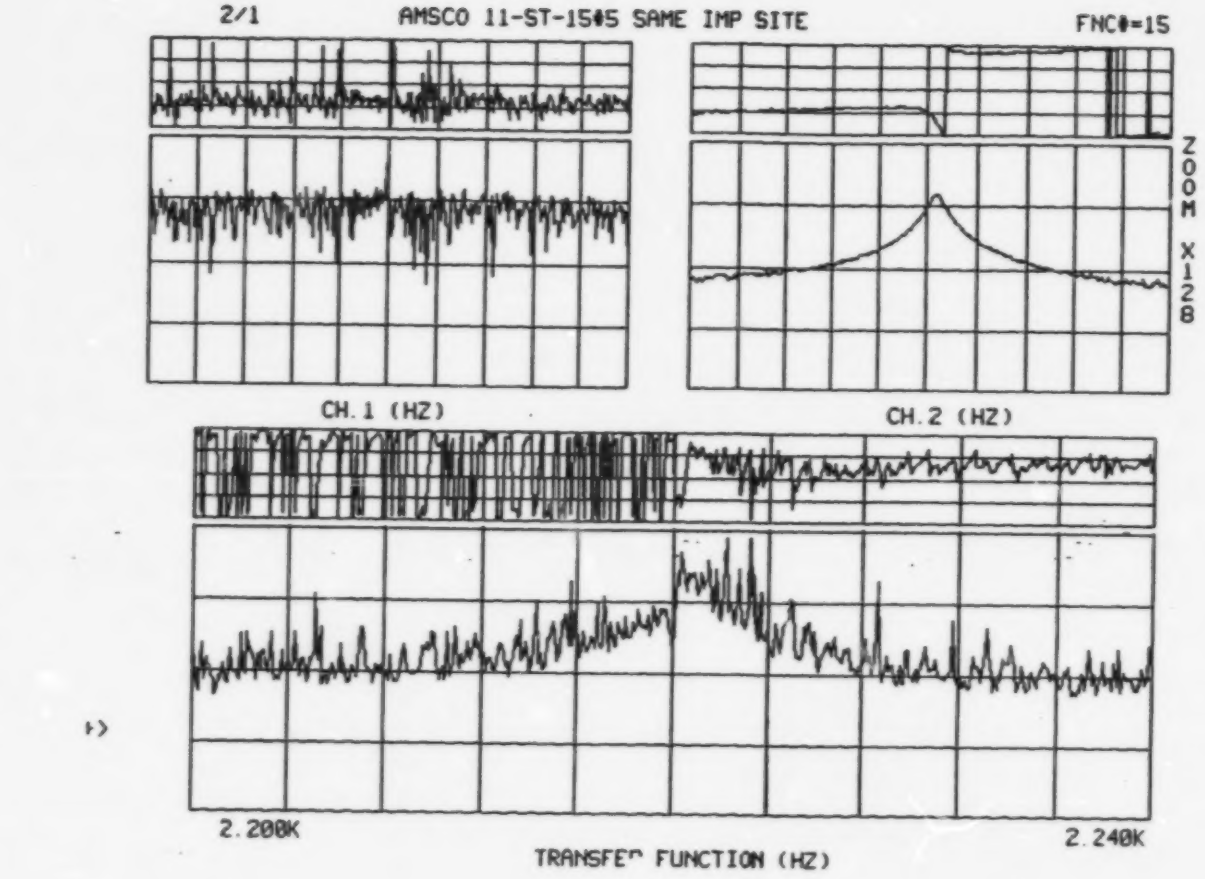


Figure 12. Impact test of a steel casting with baseband analysis frequency of 4.0 kHz and zoom factor of 128; impact force spectrum (upper left), acceleration response spectrum (lower right), and FRF (bottom).

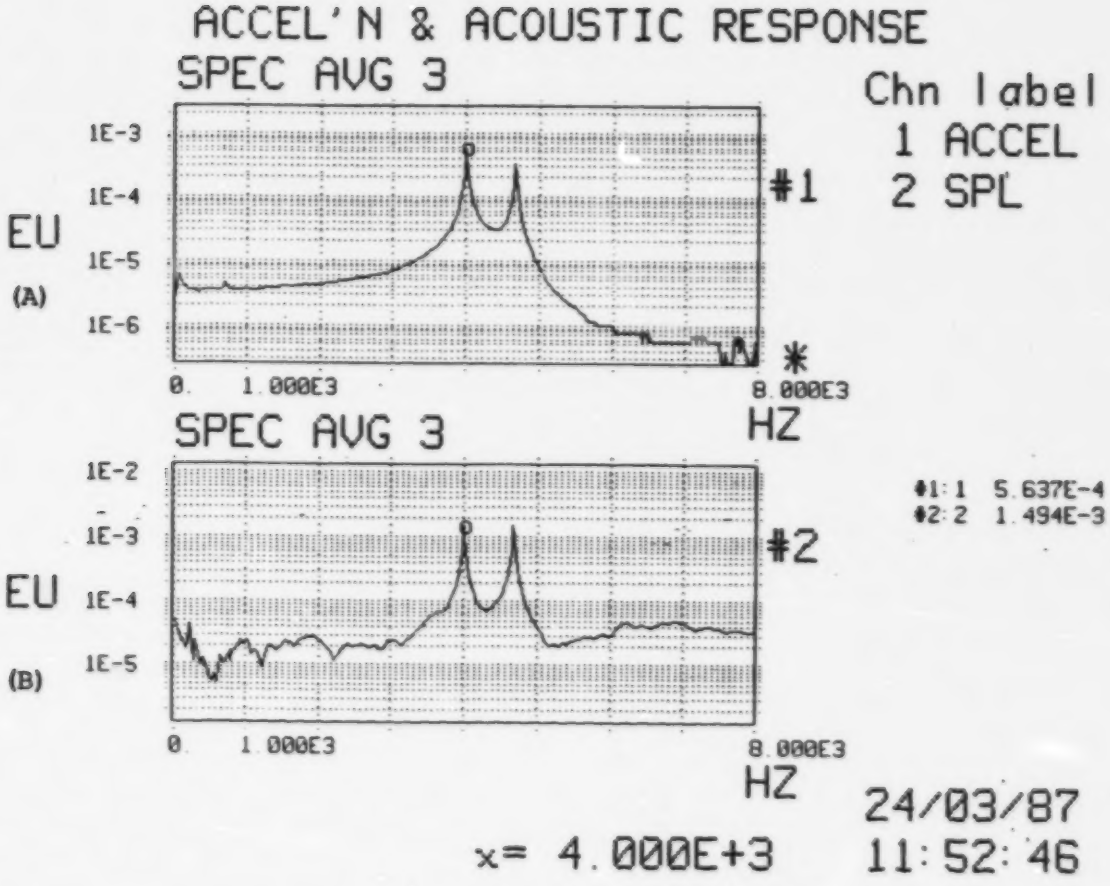


Figure 13. (a) Impact response spectrum measured on a casting using an accelerometer.
(b) Impact response spectrum of the same casting measured by a microphone in the near field.

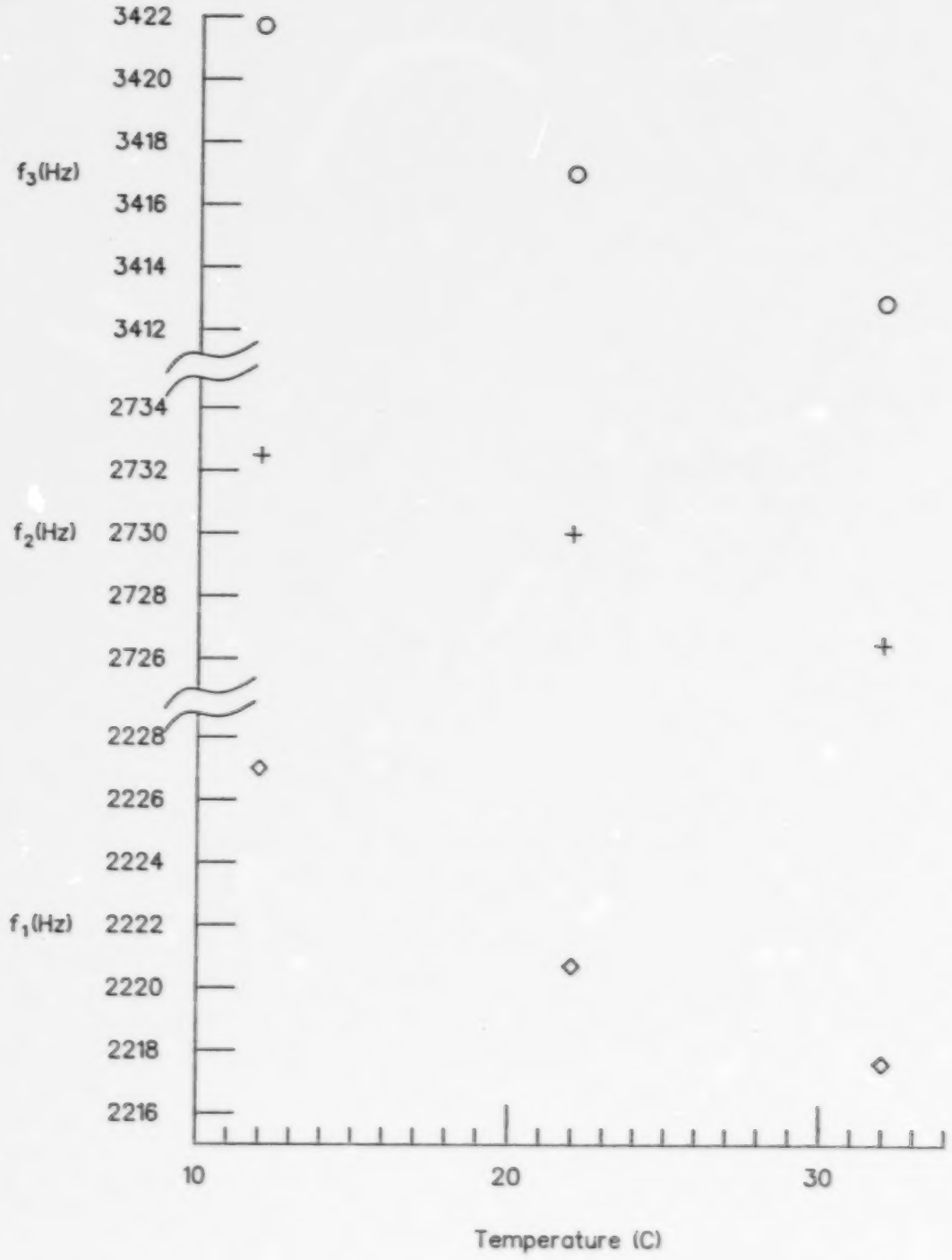
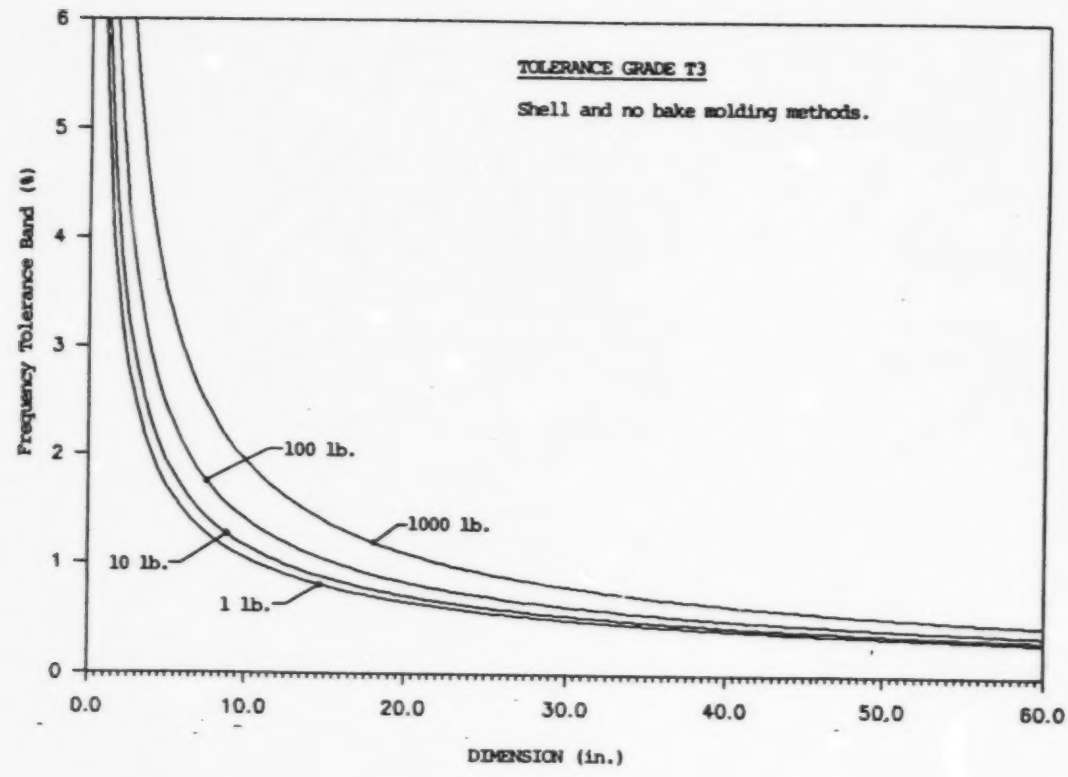
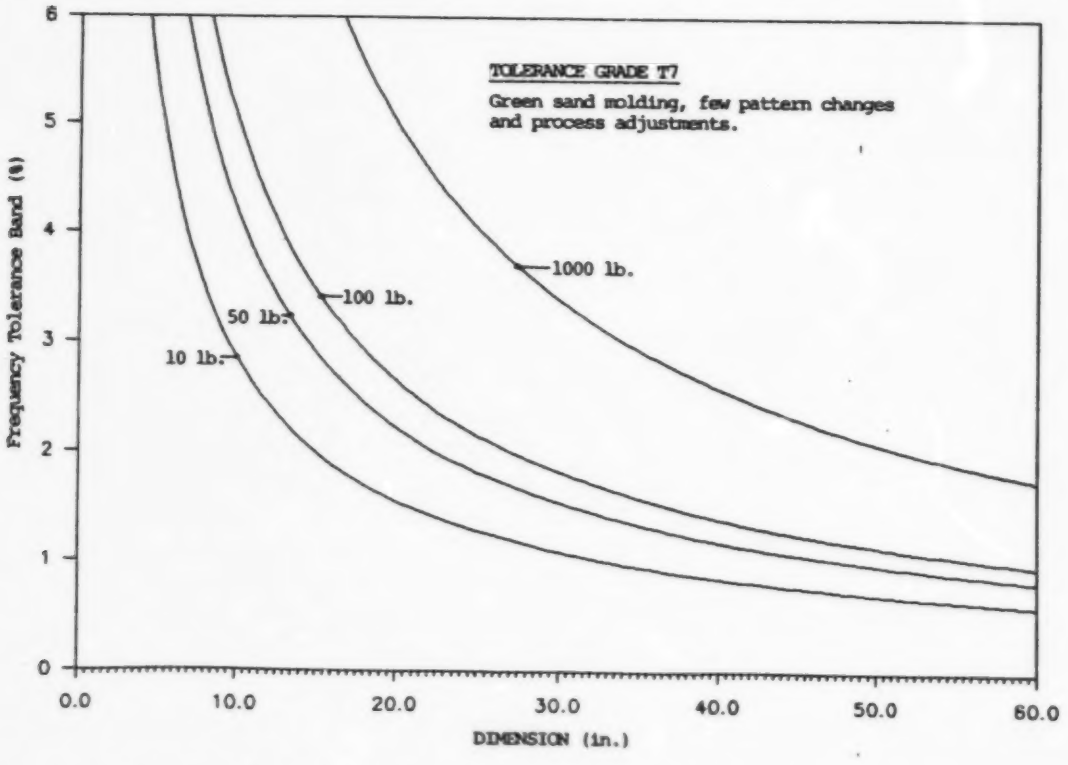


Figure 14: Graph showing the effect of ambient temperature on three natural frequencies of vibration of a steel casting



(A)



(B)

Figure 15: Graphs of relative change in natural frequency of axial vibration of uniform bars and flexural vibration of uniform beams versus dimension and casting weight for tolerance grades T3 (A) and T7 (B).

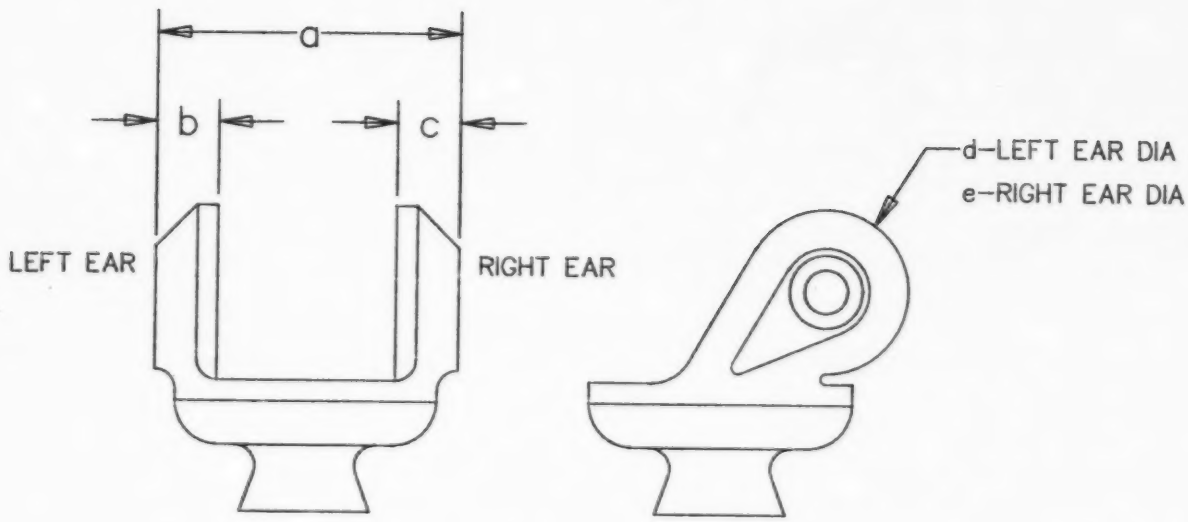
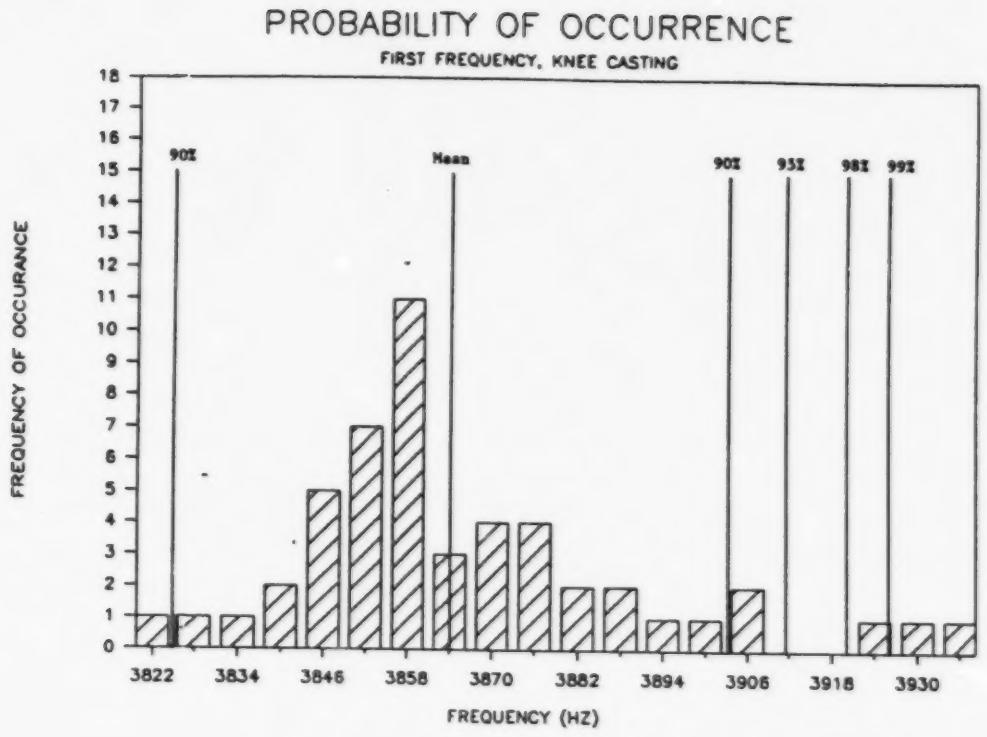


Figure 16: Artificial Knee Joint Casting


PROBABILITY OF OCCURRENCE
FIRST FREQUENCY, KNEE CASTING
FREQUENCY OF OCCURANCE
90%
Mean
90%
95%
98%
99%
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
3822
3834
3846
3858
3870
3882
3894
3906
3918
3930
FREQUENCY (HZ)

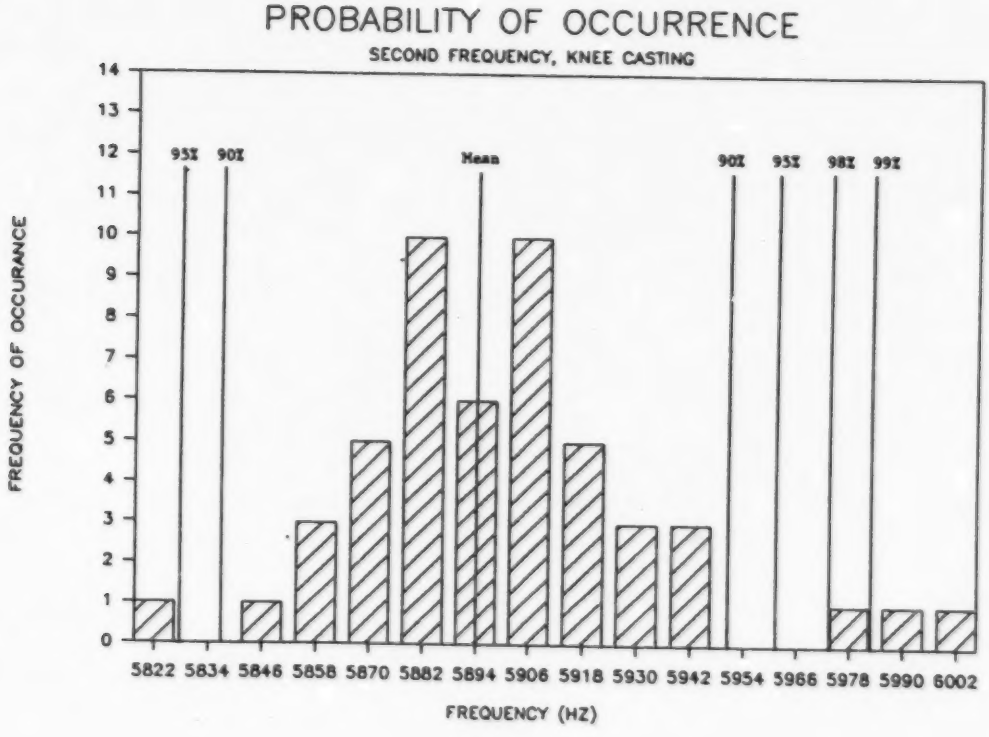

PROBABILITY OF OCCURRENCE
SECOND FREQUENCY, KNEE CASTING
FREQUENCY OF OCCURANCE
95%
90%
Mean
90%
95%
98%
99%
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
5822
5834
5846
5858
5870
5882
5894
5906
5918
5930
5942
5954
5966
5978
5990
6002
FREQUENCY (HZ)

FIGURE 17

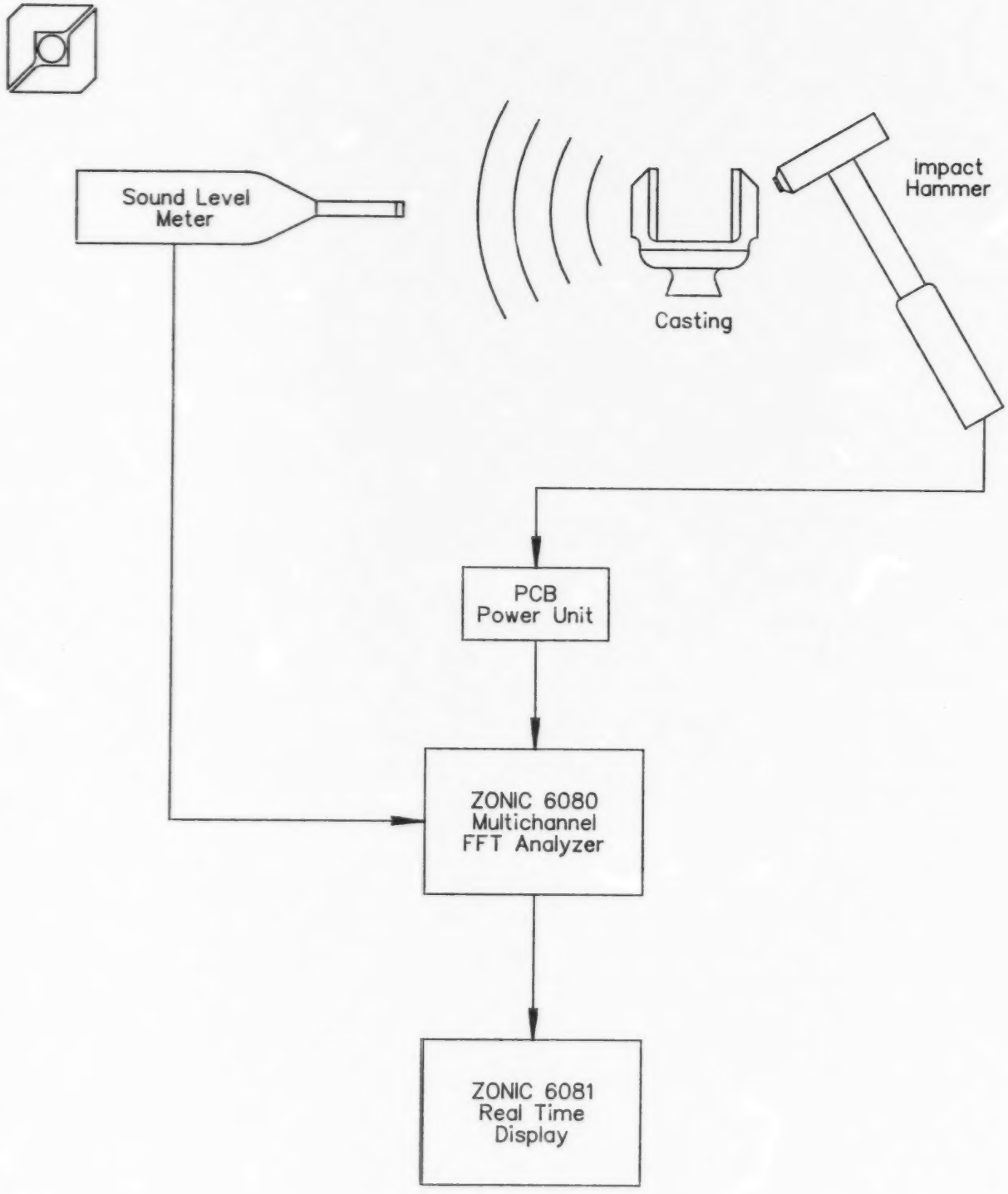
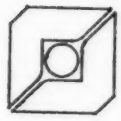


Figure 18: Freguency Analysis Test Set Up



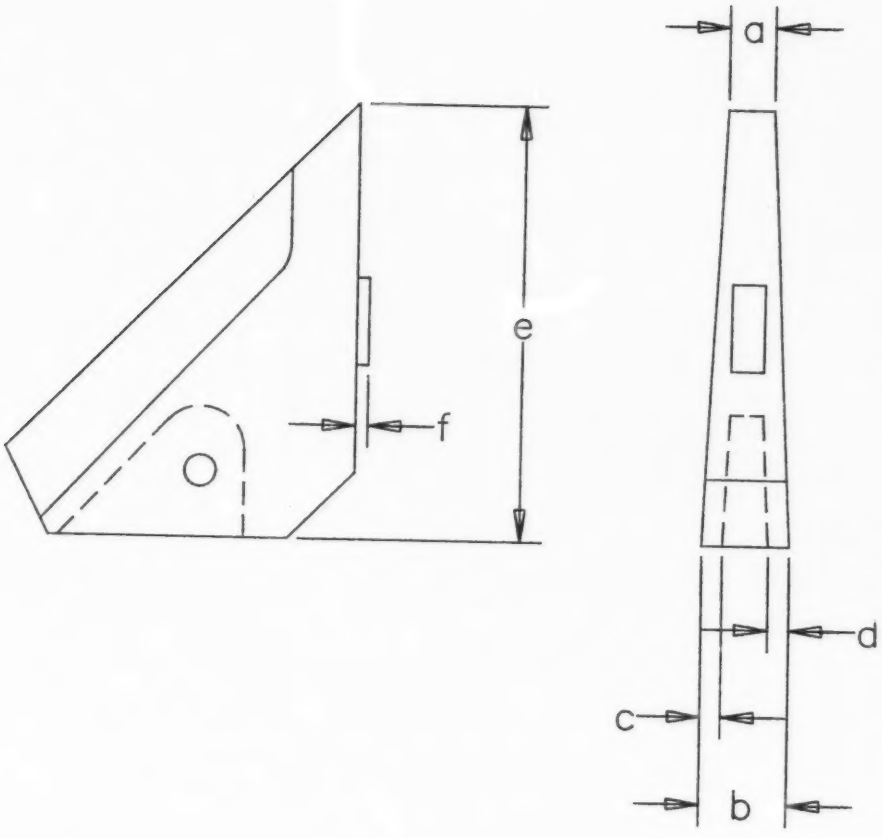


Figure 19: Chisel Plow Tooth Casting

Figure 20 [a]: Level I Defect

Figure 20 [b]: Level II Defect

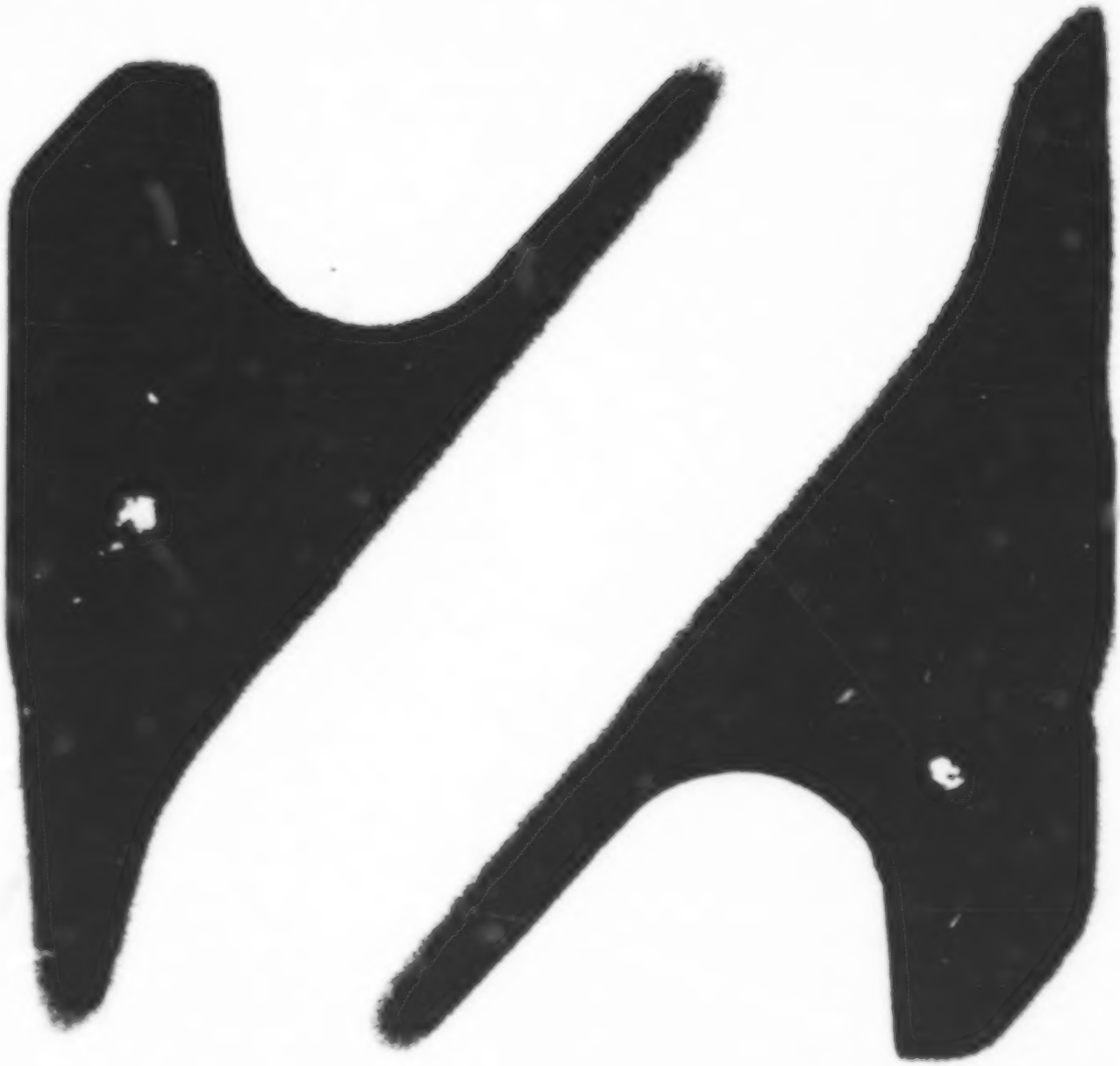
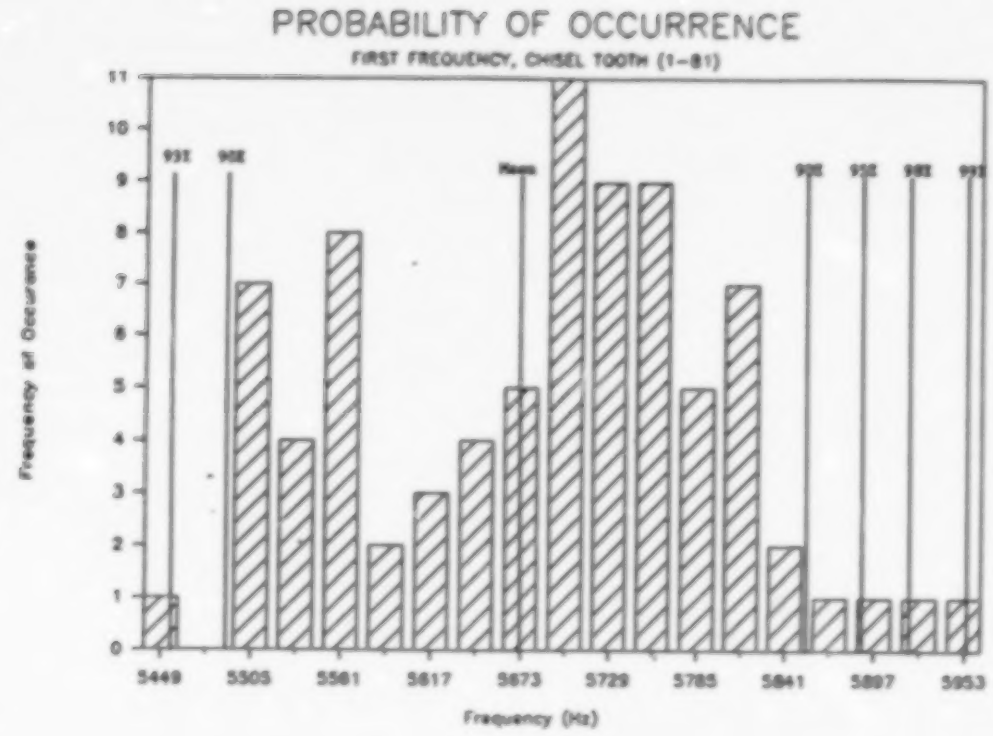


Figure 20 [c]: Level III Defect

Figure 20 [d]: Level IV Defect


PROBABILITY OF OCCURRENCE
FIRST FREQUENCY, CHISEL TOOTH (1-81)
Frequency of Occurance
Frequency (Hz)
5449
5505
5561
5617
5673
5729
5785
5841
5897
5953

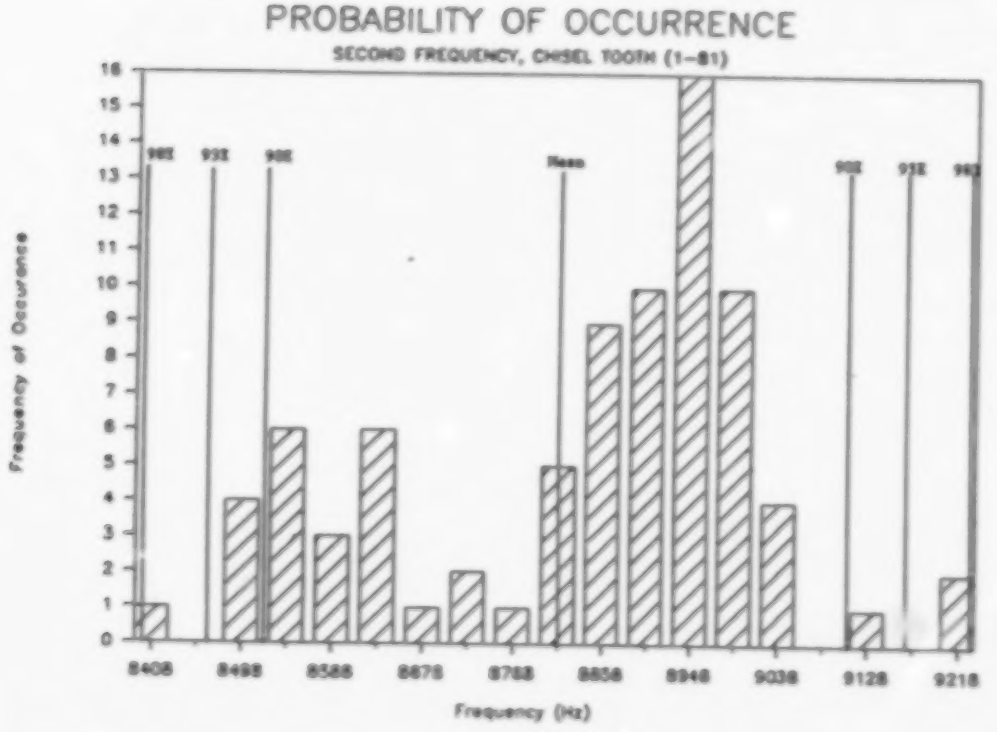

PROBABILITY OF OCCURRENCE
SECOND FREQUENCY, CHISEL TOOTH (1-81)
Frequency of Occurance
Frequency (Hz)
8408
8498
8588
8678
8768
8858
8948
9038
9128
9218

FIGURE 21

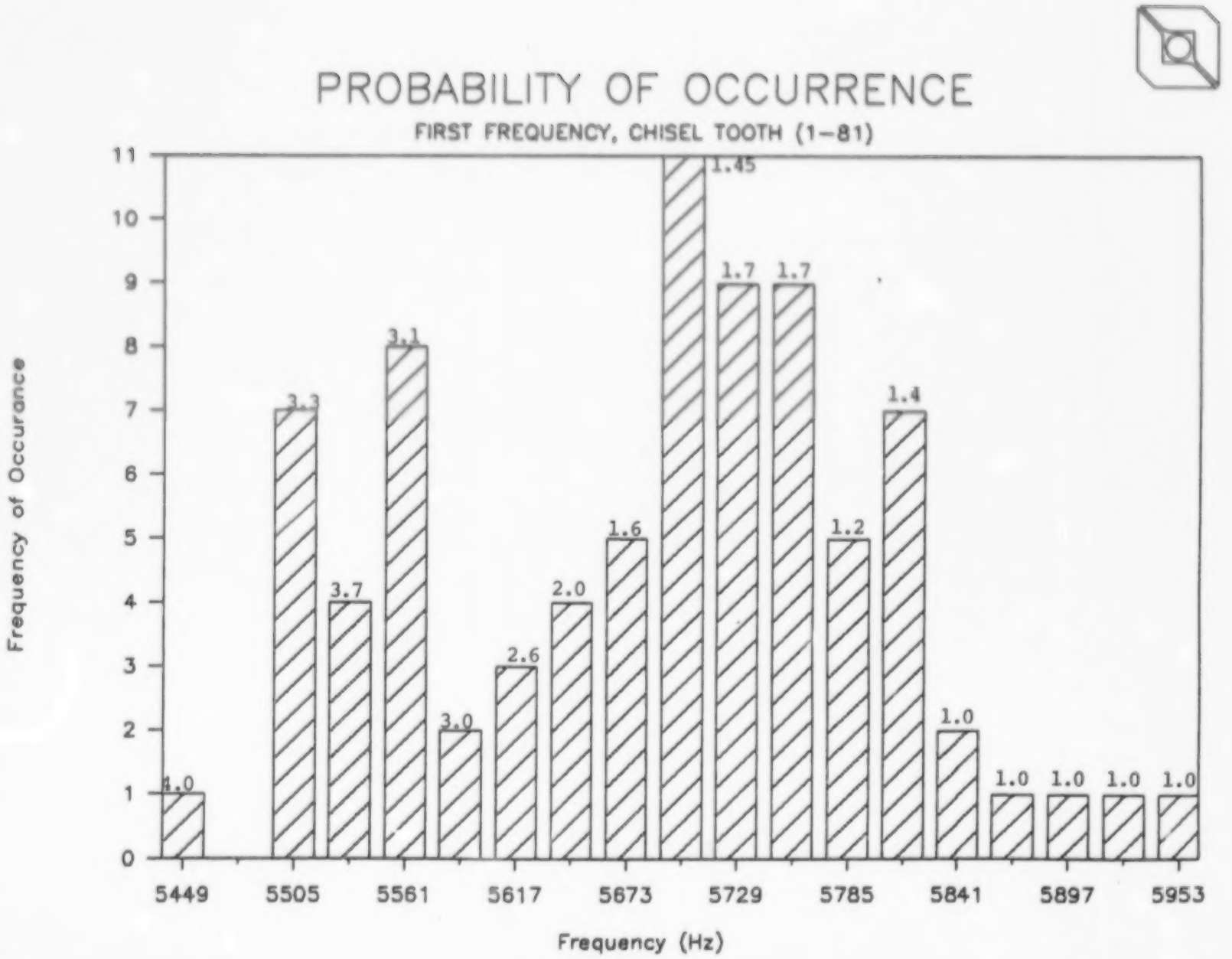

PROBABILITY OF OCCURRENCE
FIRST FREQUENCY, CHISEL TOOTH (1-81)
Frequency of Occurance
Frequency (Hz)
0
1
2
3
4
5
6
7
8
9
10
11
5449
5505
5561
5617
5673
5729
5785
5841
5897
5953
1.0
3.3
3.7
3.1
3.0
2.6
2.0
1.6
1.45
1.7
1.7
1.2
1.4
1.0
1.0
1.0
1.0
1.0

FIGURE 22

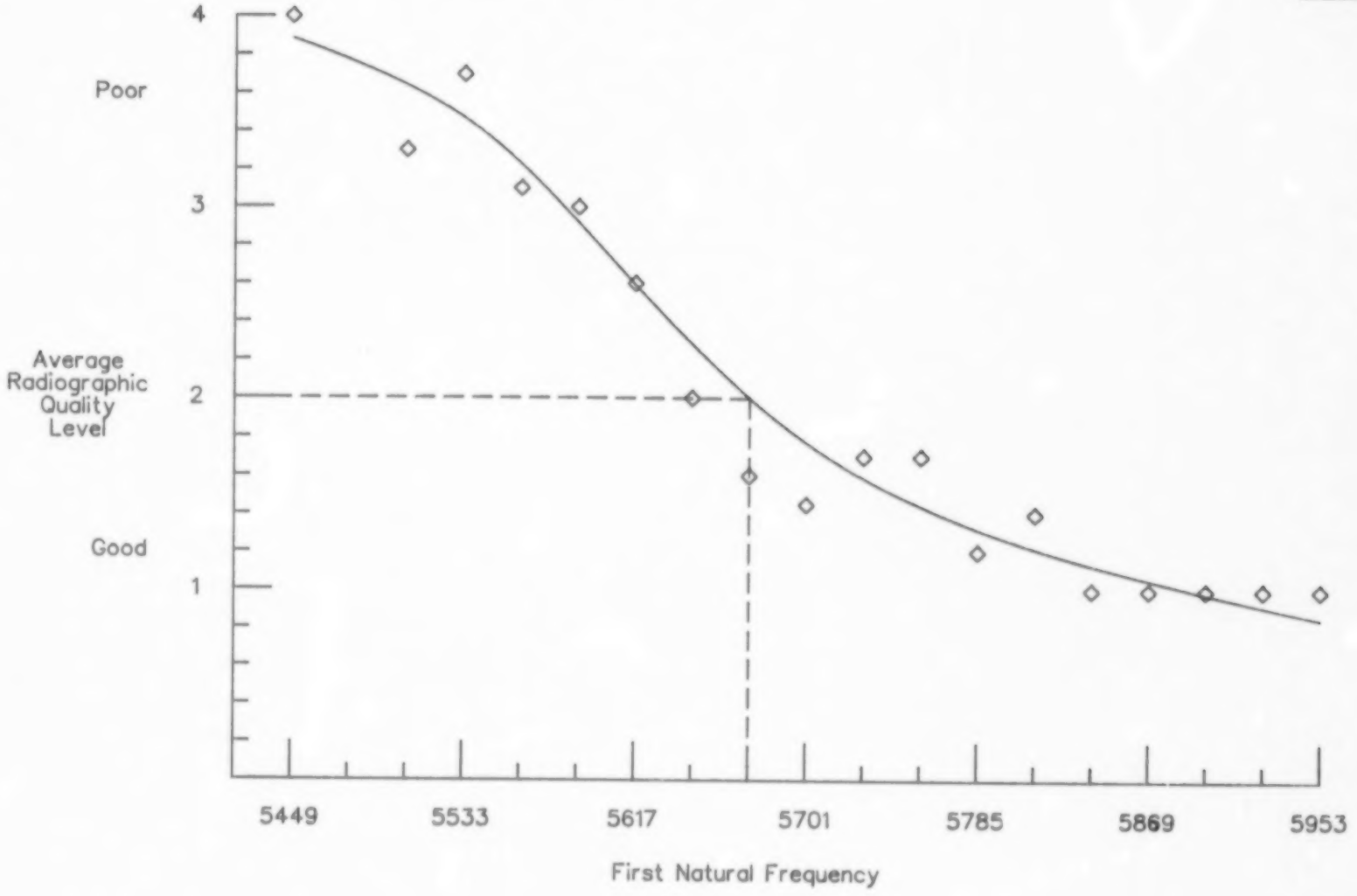

4
Poor
3
Average
Radiographic
Quality
Level
2
Good
1
5449
5533
5617
5701
5785
5869
5953
First Natural Frequency

Figure 23

Appendix

Report on NDT of knee-joint casting.

west-can INSPECTION LIMITED

Inspection Services
Industrial Radiography (Mobile Units)
X-Ray & Gamma Ray
Aircraft X-Ray (Structural)
Ultrasonic Flaw Detection etc
Thickness Gauging
Corrosion Surveys
Magnetic Particle & Penetrant
Dry & Flourescent
Visual Inspection — Buildings
Bridges, Piping, Tanks, etc
Built Up Roof Inspection

January 29, 1987

CLIENT: Industrial Technology Centre,
Manitoba Research Council,
1329 Niakwa Road,
WINNIPEG, Manitoba. R2J 3T4

CONTRACTOR: Client

Attention: Mr. Kedar Tandon

REPORT NO.	CLIENTS ORDER NUMBER	OUR INSPECTION NUMBER	INSPECTOR
1 & FINAL	VERBAL	W87 - 122-NDT	C. HENDERSON

THIS REPORT REFERS TO MAGNETIC PARTICLE & RADIOGRAPHIC EXAMINATION OF THE FOLLOWING

4 KNEE JOINT CASTINGS. PROJECT NO. 01377
IDENTIFIED 31, 35, 38 and 44

Our inspection was carried out on January 29, 1987.

The castings were radiographed and displayed no evidence of internal discontinuities.

The castings were also tested by the Fluorescent Magnetic Particle method and displayed no evidence of surface cracking or other detectable discontinuities.

WEST-CAN INSPECTION LIMITED.

Larry R Russell

L. R. RUSSELL,
OPERATIONS MANAGER,
TECHNICAL DIVISION.

LRR/br